Diesel and Gas Engine Power Plants

BY

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DIESEL AND GAS ENGINE POWER PLANTS

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Preface

This book is written primarily for the designer and operator of diesel- and gas-engine power plants, although some of the material has been used in lectures before engineering students. It is entirely practical in treatment, and the material has been used for many years by the author in his professional work as a consulting engineer.

The book deals with the plant as an entity and is not devoted solely to a discussion of engines. In fact, some readers may be disappointed because only a single chapter concerns engines. The text embraces far more than a description of engines and is in reality a working manual for the plant designer. It contains much of value for the plant operator and for engineering students interested in internal-combustion-engine power plants.

Much of the material presented is taken from the author's personal notes made over the past 12 years. It represents, therefore, the gleanings from a wide acquaintanceship with engine designers, plant operators, representatives of the oil industry, and other engineers interested in internal-combustion-engine power plants. Experience gained in the design of plants, examination of plants designed by others, and the testing of engines both in the factory and in the field has contributed greatly to the scope of the material presented.

Many nomographic charts and short-cut procedures have been incorporated into the text, primarily because they aid in simplifying many calculations. Equations are presented, and derivations of equations are used where they can serve to clarify the material being presented. Involved mathematical treatment has purposely been eliminated because too often such mathematics confuses rather than aids in a work of this character.

Although the writing of a book such as this is the work of a single individual, no one person can rightfully claim the credit for all the material presented. Many individuals, equipment manufacturers, and publications have furnished material neces-

sary to complete portions of the text. Those sources which have been used as reference or which have provided quotations used in the text have been acknowledged by appropriate footnotes. Drawings and pictures generously supplied by equipment manufacturers, technical publications, and engineering organizations have been credited to the organization supplying them.

The author wishes to thank members of the staff of the Burns & McDonnell Engineering Company for their generous assistance in assembling material for Chaps. VII, VIII, XIV, XVIII, and XIX. The author desires to express his appreciation for the assistance of L. B. Immele of the Burns & McDonnell organization in developing the nomographs for Figs. 50 and 51.

Power has graciously extended to the author permission to reproduce many exhibits originally appearing in that magazine, as well as permission to use material in Chaps. XII and XV which appeared as articles in *Power*.

The author is indebted to C. E. Beck, J. H. Gallaway, L. N. Rowley, Jr., C. W. Good, W. L. H. Doyle, and Charles Keane for their encouragement, suggestions, and criticism in connection with the preparation of the manuscript. To his wife must go the credit for correcting and proofreading the manuscript and the final printed book.

GLENN C. BOYER.

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DIESEL- AND GAS-ENGINE POWER PLANTS

CHAPTER I

GENERAL INTRODUCTION

When it is realized that over 90 per cent of the total horsepower of prime movers in the United States consists of internal-combustion engines, it is not surprising that this type of prime mover is being used extensively in central stations and industrial power plants. Although the bulk of the internal-combustion-engine horsepower is employed in motor-vehicle and agricultural service, there is a substantial amount employed in stationary service.

1. Engines for Stationary Service.—Most engines used for stationary power service are of the compression-ignition type in which the heat for igniting the fuel charge results from the compression of the air in the engine cylinder before introduction of the fuel. This type of engine is popularly termed a *diesel engine* in the United States in honor of one of the men responsible for its creation and early development. Oil is used for fuel in most engines, although natural gas is also employed. Recent developments include engines capable of burning either liquid or gas fuel.

The pioneering in the adaptation of internal-combustion engines to central-station and industrial-power service has been done, and today this type of prime mover is an important element in the field of small and medium-sized power plants as indicated by the widespread use in municipal and private power plants, pipe-line pumping stations, manufacturing and industrial establishments, and apartment-building and institutional power plants.

2. Central Stations.—A recent study of generating stations made by the Federal Power Commission covering plants having an installed capacity of 1,000 kw or more per station gave the

following installation of diesel engines during the past four decades.

The largest plant now installed in the United States is at Vernon, Calif., while the second largest is at Brawley, Calif.,

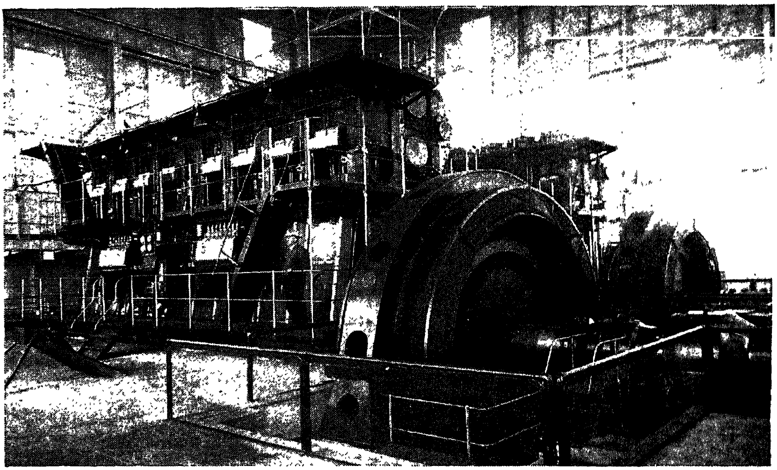

FIG. 1.—Diesel plant of the Salt River Valley Water Users' Association, Tempe, Ariz., containing two 7,000-bhp two-stroke-cycle double-acting Hamilton engines. (*Courtesy of General Machinery Corporation.*)

in the Imperial Valley. A summary of the nine largest diesel operating plants in the United States is contained in Table 2.

TABLE 1.—DIESEL GENERATING CAPACITY IN CENTRAL STATIONS

Period	Private, kw	Municipal, kw	Total, kw
Before 1910	520	0	520
1910–1919	11,150	3,858	15,008
1920–1929	94,398	82,930	177,328
1930–1939	100,792	211,663	312,455
Total	206,860	298,451	505,311

3. Evolution of the Small Power Plant.—Thirty-five years ago, the small community or industrial establishment that desired the advantage of electric service had to build a plant using steam

TABLE 2.—LARGEST DIESEL-ELECTRIC GENERATING PLANTS IN THE UNITED STATES

Location	Installed hp	Number of units
Vernon, Calif	35,000	5
Brawley, Calif	18,340	8
Tucson, Ariz	15,290	9
Tempe, Ariz	14,000	2
Rockville Center, N. Y	12,500	6
Rushville, Ind	12,240[a]	4
Freeport, N. Y	11,900	7
Cardin, Okla	10,750	4
Ponca City, Okla	7,250	5

[a] Three units installed, fourth unit on order.

engines as prime movers. The improvements in the arts of both generation and transmission of electrical energy improved constantly, and the early twenties saw the promulgation of the plan for "superpower" in which huge generating stations connected by networks of transmission lines would blanket the United States. Many large steam-electric and hydroelectric plants have been and are still being built with their attendant high-voltage transmission lines conveying large blocks of electrical energy over immense distances. Remarkable economies in the production of electrical energy have resulted from the construction of high-pressure and high-temperature steam-electric stations, containing units of great capacity. However, the cost of transmitting the energy produced in these stations over long distances has in many cases largely absorbed the savings made in the cost of generation.

While the basic idea behind the superpower program was sound, three factors have worked against the full realization of the advantages claimed for it. They are:

1. Inability to operate superpower stations at most economical loads in many instances.

2. Transmission difficulties due to weather conditions and electrical characteristics of generation and transmission facilities.

3. Relative cost at a given locality of power delivered by a transmission line or generated in a diesel-electric plant where required.

Flexibility of plant operation is a vital necessity in all electric generating stations, as has been proved by the experience of the electric utilities during the past decade. Base-load plants operating at full capacity and producing a constant quantity of electrical energy every 24-hr period can and do show remarkable economies. If people all used the same amount of electrical energy each hour of the day, there would be no peak-load difficulties and many of the problems now confronting the power-plant designer and operator would not exist.

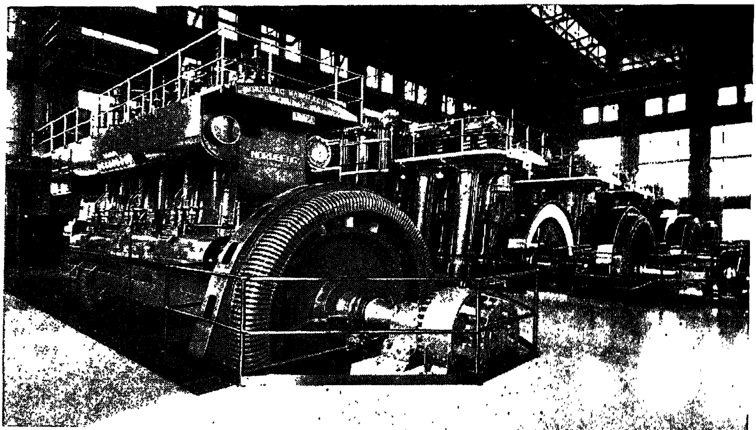

FIG. 2.—Municipal power plant at Ponca City, Okla., containing three 1,250-hp and one 2,250-hp two-stroke-cycle diesel engines. (*Courtesy of Nordberg Manufacturing Company.*)

Since human nature still impels individuals to attempt the satisfaction of their own desires without regard to other individuals and their wants, and since one of the desires of most individuals is the use of electricity for light and other purposes at any hour of the day or night, those supplying the electrical energy required will always be confronted with the necessity of meeting the heavy as well as the light demands for electricity.

4. Power-transmission Problems.—Transmission-line difficulties and resulting power outages caused by the caprices of Dame Nature are well known to electrical consumers and power-system operators alike. Charles F. Steinmetz's famous comment that lightning seldom strikes twice in the same place because there

generally is not anything to hit the second time has more than wit behind it. Many electric utilities operating extensive transmission systems think nothing of one to three outages daily somewhere on their systems from lightning disturbances during certain months of the year.

Aside from possible outages due to lightning, tornadoes, hurricanes, sleet, or other weather stresses beyond the mechanical and electrical capacities of the transmission facilities, long transmission lines are subject to electrical difficulties including switching surges, synchronizing difficulties when generating stations are connected by a long transmission line, frequency variation on extensive systems with several generating stations connected, as well as electrical difficulties due to vagaries of relay operation caused by abnormal power-factor conditions. Furthermore, long-distance transmission of electrical energy is not economically sound, as has been shown.[1]

A major portion of these difficulties disappears when the electric generating station is brought close to the load it must supply. For example, a generating station in the basement of a hotel supplying only that institution is not subject to lightning disturbances, nor do transmission-line failures of any description influence the electrical service. The only sources of trouble are within the generating equipment or the building wiring, and the latter source of trouble is present irrespective of the source of electricity.

5. Economy of Diesel Engine.—Operating reports of electric generating stations using diesel engines for prime movers contained in the annual Report on Oil-engine Power Cost of the American Society of Mechanical Engineers are ample proof that, in many cases, the small diesel power plant can produce electricity for an industrial establishment, water-works system, municipality, or other consumer cheaper than it can be delivered by other means, be it a superpower system or an electric utility using either steam or hydro as a source of power.

Since the engineer is interested in securing electrical energy at the lowest possible cost, many occasions occur when it is necessary for him to give serious consideration to this form of prime mover which is coming more into prominence every year.

[1] SPORN, PHILIP, Cost of Generation of Electrical Energy, *Proc., A.S.C.E.*, vol. 63, No. 10, p. 1932, December, 1937.

After a lapse of many years, the small electric power plant is again coming into its own with diesel and gas engines as prime movers simply because it can produce electrical energy in many instances as cheaply and as reliably as any other prime mover. It is coming back through the development and perfection of the compression-ignition engine and the abundance of cheap fuel oil and gas.

6. Preliminary Study Needed.—Since the present-day diesel engine offers an attractive possibility as a prime mover, and since

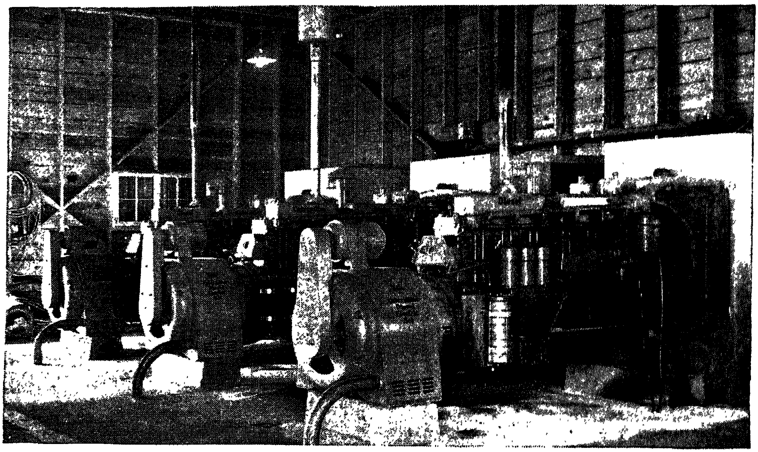

FIG. 3.—Three Caterpillar diesel-electric sets furnish power for a Bentonite mining plant. (*Courtesy of Caterpillar Tractor Company.*)

the engineer in studying any problem involving the procurement of primary power should not overlook any possible economical source, consideration should be given to diesel engines wherever the cost of fuel oil or gas is such that the resulting cost of producing power with a diesel engine is equal to or less than the cost of producing power with steam prime movers or purchased electrical energy.

In order to assure himself that a diesel engine should be installed, it is necessary that the engineer, in making a study of a power problem, weigh the operating and fixed charges on a diesel plant along with the possible operating and fixed charges when using steam-driven equipment, or when considering the purchase

of electrical energy wholesale. Each problem of power supply must be considered on its own merits, and general theorems have no place when a particular problem is under consideration. It is necessary, therefore, in selecting any power source that the one making the study have at his command all the basic information with which to analyze the problem properly in order that the most economical power source be made available.

7. Factors to Consider.—Any preliminary study of a problem involving the production of power and the selection of the most economical method for its production involves consideration of the following factors:

1. Nature and character of load to be supplied.
2. Size of units required for supplying power.
3. Space available or required for housing units.
4. Investment necessary in power-generating facilities and accessories.
5. Cost and character of fuel available.
6. Cost and character of labor necessary.
7. Quantity and character of available cooling water.
8. Maintenance costs.
9. Auxiliary power requirements.

With this information, it will be possible to develop the operating and total costs for power produced either by diesel or steam prime movers. These costs can then be compared with each other or with the cost of purchased power (if purchased power is under consideration) in order to arrive at the cost of power supplied by any one of the several sources contemplated.

It should always be borne in mind that one is seeking to obtain the lowest cost consistent with reliability. Cheap cost of power without reliability may be the costliest power. Reliability, for reliability's sake only, may on the other hand be the most expensive of luxuries.

CHAPTER II

VARIABLE LOADS IN POWER PLANTS

A few years ago a newspaper acquaintance remarked, during a conversation, that the most constant thing in this world was change. While his remark was based upon his experiences in contact with the daily ebb and flow of events that we designate as *news*, it very aptly describes a host of problems facing many engineers, particularly those who deal with problems involving the use of power in any of its various forms.

The demand for electrical energy imposed upon a central station, for example, is continually varying from hour to hour, day to day, and year to year. This same variable demand is experienced by the water-works pumping station, a power plant serving an industrial establishment, and in many other places where power is produced or consumed.

8. Daily Load Curves.—Since the demand for electrical energy at a generating station is continually varying, it is necessary for us to have some means of portraying this variation in such a manner that it becomes intelligible. Daily-load curves, such as those shown in Fig. 4, are used extensively to show the relation between load and time for a single day or for several days as a means of rough comparison.

On this curve, the horizontal axis or *abscissa* is time and the vertical axis or *ordinate* is load in kilowatts. Time starts from midnight and ends at midnight so that the chart portrays the variation in load for the entire day. An inspection of the two curves on Fig. 4, which are taken from actual plant operating records, shows a considerable variation in the load supplied by the station.

Daily-load curves are valuable in assisting the plant operating staff to estimate load changes on the days to follow, determine shift in the time at which the maximum daily load occurs during the various seasons, and detect unusual load shifts which may need investigating on the part of the management.

Instruments are available for use in central stations producing electrical energy which plot this load curve on a continuous chart for the convenience and guidance of the plant staff. Such an instrument is known as a *graphic wattmeter*.

While the load curves in Fig. 4 show what happened throughout the 24 hr of two specific days, they can never give a complete

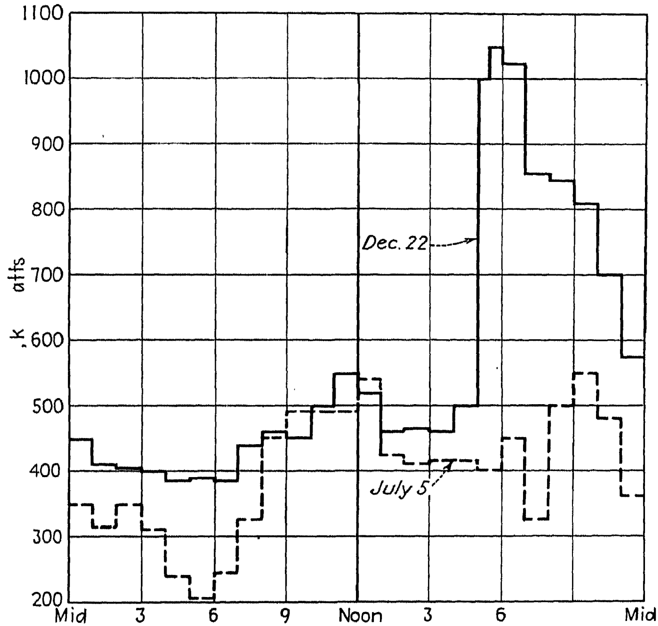

FIG. 4.—Typical daily load curves.

picture of the load variation which that plant was called upon to furnish throughout the year. In order to present the complete story of load variation, it will require not two but 365 daily-load curves. Should the attempt be made to plot all 365 curves on Fig. 4, they would interlace to the point of being meaningless.

In an attempt to present a picture of the load variation for a period of one year, it has been customary in many electric systems to cut out of heavy cardboard a load curve for each day during the year and to stack these curves in chronological order

to give a picture of the variation in load. Such an assembly of daily-load curves for one system is shown in Fig. 5.

9. Load-duration Curves.—While daily-load curves are of value in showing the variation in load from hour to hour over a

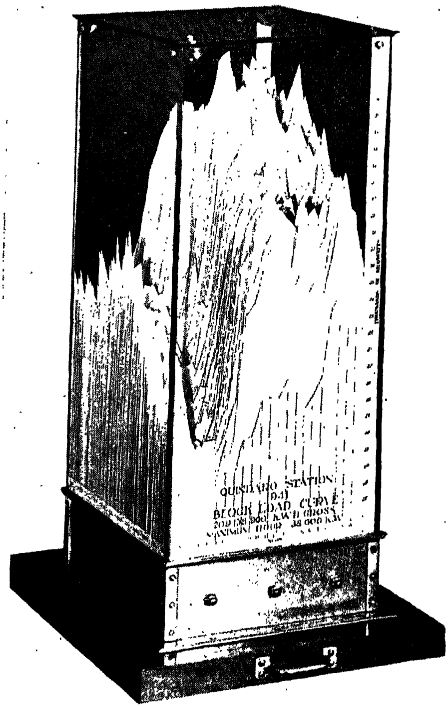

FIG. 5.—Block load assembly showing variation in daily electrical load output. (*Courtesy of Board of Public Utilities, Kansas City, Kansas.*)

definite 24-hr period, it is also necessary to have a means of presenting the load variation for a total of 8,760 hr, or one year, in such a manner that the data will be of value. The load-duration curve, of which Fig. 6 is an example, is the best known means for presenting the variation in load for any period of time in a manner that lends itself to ready interpretation. As plotted in Fig. 6,

any point on this curve shows the percentage of the total hours in the year when the load exceeded a given value. Thus, while the peak load on the plant was 1,050 kw, the load exceeded 575 kw 20 per cent of the time and 450 kw 50 per cent of the time.

In discussing Fig. 4 containing two daily-load curves, the limitations of presenting 365 daily-load curves on the same chart

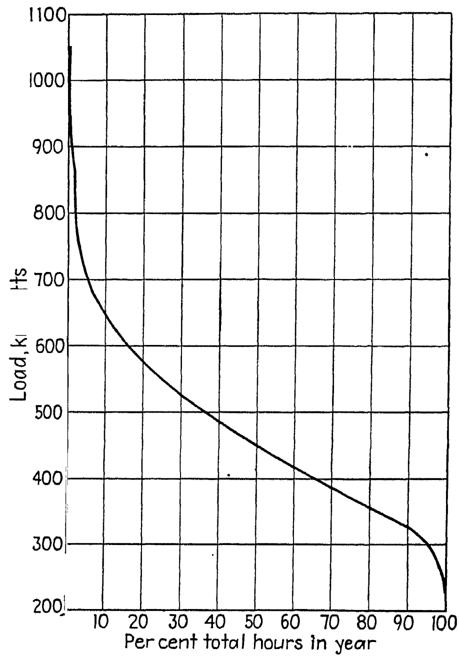

FIG. 6.—Load-duration curve.

were mentioned. Figure 6 contains the information obtained from the 365 daily-load curves, of which those in Fig. 4 are only two, in a form that is usable for studying the load variation in this particular electric generating station.

Load-duration curves are developed by grouping all the hours during a year when a particular load occurred, and then by starting with the largest value of load, hours are accumulated for all preceding larger load values. Thus for each load value

plotted on the load-duration curve, the time value corresponding to that load is the summation of all the hours at which this load as well as larger loads occurred. As an illustration of the development of the load-duration curve, the daily-load curve for Dec. 22, in Fig. 4, will be used as the data from which to evolve a load-duration curve. Table 3 summarizes the grouping of the information taken from the daily-load curve in order to convert it to a load-duration curve.

TABLE 3.—DEVELOPMENT OF LOAD-DURATION CURVE

Load, kw	Number of hours occurring	Summation, total hr
1,050	1	1
1,025	1	2
855	1	3
845	1	4
810	1	5
700	1	6
575	1	7
550	1	8
520	1	9
500	2	11
465	1	12
460	3	15
450	2	17
440	1	18
410	1	19
405	1	20
400	1	21
390	1	22
385	2	24

The values in Table 3 for the load in kilowatts and the summation of total hours are the values that are plotted in graphical form to produce the load-duration curve shown in Fig. 7.

It may appear that the construction of a load-duration curve for the electrical energy production of a power plant is a tremendous task since it involves the classification of 8,760 hourly figures. While some time is required in assembling the data in

a form similar to that given in Table 2, the process may be speeded up by grouping values within a definite load range and plotting a single point as the average of the group. Thus in producing a load-duration curve such as Fig. 6, it is sufficiently accurate to take a range of 50 kw for a single point on the curve. Thus all load values between 401 and 450 kw would be grouped together, and when plotted on the duration curve, it would be plotted as 425 kw which is a close approximation of the average of all the values in the group. The highest value in kilowatts is generally taken as the actual value, as is the point of lowest load.

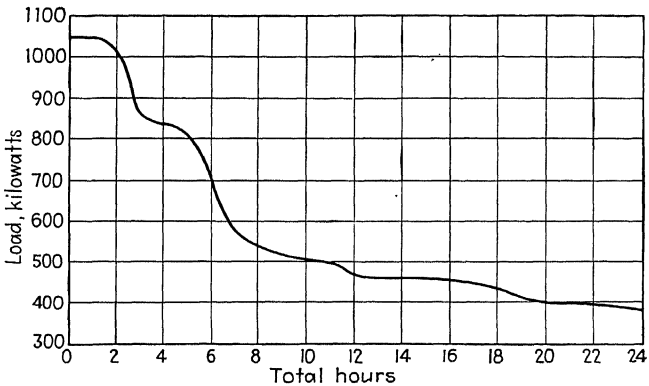

FIG. 7.—Load-duration curve developed from daily-load curve for December 22 shown in Fig. 4.

The variation in a series of load-duration curves for an actual power plant covering a period of several years is presented in Fig. 8. This series of curves indicates that, in general, the shape of the load-duration curve for a particular plant will remain fairly constant. Thus in making studies of future load conditions, probable duration curves can be estimated based upon the shape of the curve developed from the most recent plant information available and the estimates available for the probable rate of load growth.

It should be pointed out in passing that, since the abscissa or horizontal axis of the duration curve represents time and the ordinate or vertical axis represents load, the area under the curve represents power output. This property of the curve will be

considered in more detail in the following portion of this chapter. The load-duration curve can also be employed as a graphical representation of annual load factor since the ratio of the area under the duration curve to the area of the rectangle bounded by X and Y axes, the abscissa for the maximum load on the plant and the ordinate for 100 per cent total hours in the year, is annual load factor.

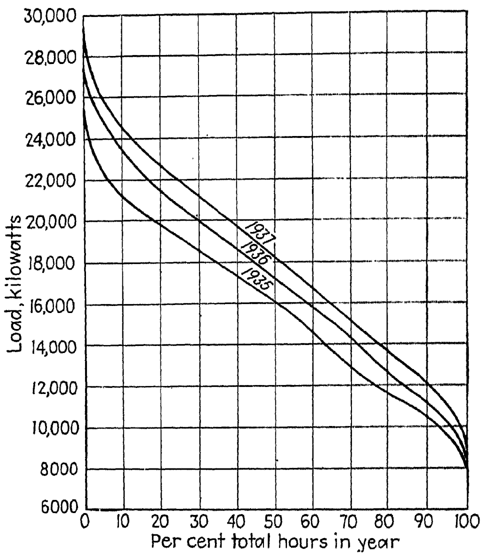

Fig. 8.—Variation in load-duration curves from year to year in a central station.

10. Analyzing Variable Loads.—In the consideration of an improvement program for the electric generating plant previously referred to in Figs. 4 and 6, it is desirable to know the kilowatt-hours generated per gallon of fuel oil during a year's period, if diesel prime movers are used. The load-duration curve in Fig. 6 was derived from actual plant operating data and will be used in the analytical method discussed here. In addition to the load-duration curve, the guaranteed performance for the diesel-engine unit or group of units under consideration is also necessary.

Let us consider for this plant the use of two diesel-engine generating units each having a full-load capacity of 870 kw. The operating schedule will require that the second unit be started when the plant load reaches 750 kw, and the plant load above this point will be divided equally between the two machines. With this operating schedule and the fuel guarantees of the units under consideration, a curve of engine performance can be plotted with

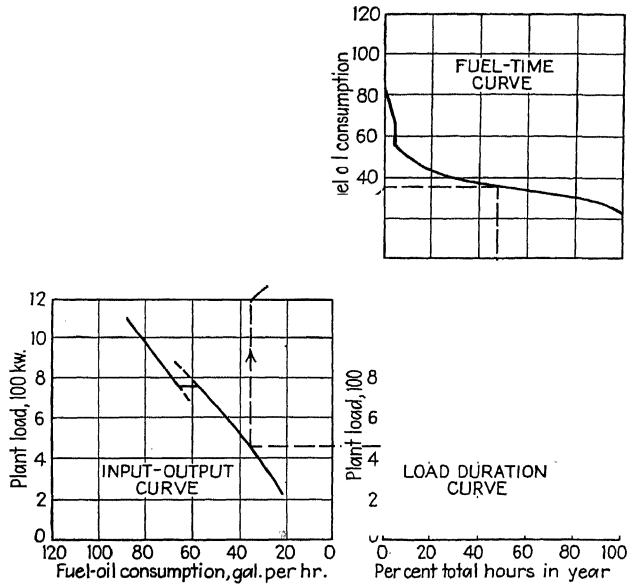

FIG. 9.—Graphical analysis of variable load in diesel-electric generating station.

plant load in kilowatts as the ordinate and gallons of fuel oil per hour as the abscissa. For conversion from engine manufacturer's guarantees of pounds of fuel consumed per kilowatt-hour to kilowatt-hours produced per gallon of fuel oil, a fuel weighing 7.29 lb per gal was used, corresponding to a fuel ranging from 28 to 30 A.P.I. gravity.

In the lower right-hand corner of Fig. 9 is reproduced the load-duration curve from Fig. 6, and in the lower left-hand corner of Fig. 9 is plotted the performance or input-output curve for the group of generating units considered. In plotting the load-

duration curve, the abscissa values increase in the right-hand direction from the origin, while in the case of the input-output curve the abscissa values increase in the left-hand direction from the origin. The values for the ordinates are common to both curves.

With the use of these two curves and elementary mechanical projection, a third or fuel-time curve is developed in the upper right-hand corner in which the abscissa is percentage of total hours and the ordinate is gallons of fuel oil per hour. The mechanics of the development of the fuel-time curve are shown by means of the dashed lines and arrows for locating one point on the curve. Fundamentally, this fuel-time curve shows the percentage of time during which the rate of burning fuel oil exceeded a certain number of gallons per hour. Since the abscissa of this fuel-time curve is time and the ordinate rate of oil consumption, the area under the curve is proportional to the quantity of fuel oil consumed in generating the quantity of electrical energy represented by the area under the load-duration curve. This proportion is readily obtained. If, for example, 1 in. horizontally represents 20 per cent of the total hours in the year, or 1,752, and 1 in. vertically represents 10 gal of fuel oil per hour, then 1 sq in. of area under the fuel-time curve represents 17,520 gal of fuel oil consumed.

For office use, the three curves on Fig. 9 are plotted on a sheet of cross-section paper approximately 17 by 20 in. Since standard cross-section paper divided 10 by 10 to the inch is available in rolls 20 in. wide, no difficulty is experienced in obtaining sheets of practically any dimensions for developing the curves in Fig. 9.

Areas under the load-duration and fuel-time curves can be obtained either by the use of a planimeter or by counting squares. From the curves developed with this analysis in Fig. 9, the following information was obtained.

TABLE 4.—SUMMARY OF DIESEL-PLANT ANALYSIS

Total energy generated (actual kilowatt-hours)	3,854,400
Total energy generated (area under curve, kilowatt-hours)	4,150,000
Per cent error	7.5
Total fuel-oil consumed (area under curve, gallons)	351,000
Kilowatt-hours generated per gallon fuel oil $\frac{4,150,000}{351,000}$	11.85
Btu per kilowatt-hour generated	11,800
Over-all plant thermal efficiency, per cent	29

It is seen from Table 4 that an error of 7.5 per cent is introduced in development of and determination of the area under the load-duration curve. For analytical purposes, all calculations are based upon the areas found under the various curves, owing to the fact that an error introduced into the load-duration curve is automatically reflected in the fuel-time curve. By using the values determined from the curves, the percentage error in the value obtained for kilowatt-hours generated per gallon of fuel oil is less than if the actual kilowatt-hours generated and the value for the total fuel oil consumed as determined from the fuel-time curve are employed. By using the latter combination of values, a 7.5 per cent error is automatically introduced into the value for the quantity of fuel oil consumed, without any compensating error in the quantity for total generation in kilowatt-hours.

The input-output curves for contemplated improvements employing other sizes of generating units could be plotted on the lower left-hand portion of Fig. 9 and their respective fuel-time curves developed. These would enable the designing engineer to select readily the most economical combination of units from the fuel-economy standpoint.

11. Willans Line Method.—In many instances, the quantity of oil required by a diesel engine for producing a variable output can be determined more rapidly than by the method previously outlined. This short cut is based upon the fact that the fuel-oil consumption of most modern diesel engines tends to follow a Willans line. The Willans line, originally developed for showing the performance of steam engines and turbines, is a curve in which total fuel input per hour is plotted against load on the unit. For steam engines and turbines this curve is practically a straight line from one-fourth to full load on the unit. Likewise with diesel engines, from one-half to full load the curve of total fuel consumed per hour is practically a straight line when plotted against load on the unit.

Owing to the fact that for all practical purposes this curve remains straight from full load down to approximately 40 per cent load, a point on this curve representing the average load on the diesel-engine generating unit will likewise give the average quantity of fuel oil consumed by the unit in pounds per hour, provided that the minimum load carried by the unit is not less than about

35 per cent of the capacity of the unit. *This is true irrespective of how the load supplied by the diesel generating unit might vary.*

Consider, for example, a diesel-engine generating unit of 1,500-kw capacity generating 8,760,000 kwhr annually and having fuel-oil consumption as shown in the following table.

TABLE 5.—FUEL-OIL CONSUMPTION BY DIESEL ENGINES

Load, kilowatts	1,500
Pounds oil per kilowatt-hour generated	0.598
Total pounds oil used per hour	

These fuel-consumption data are plotted in
Willans line. Now since the average load on the unit is 8,760,000/8,

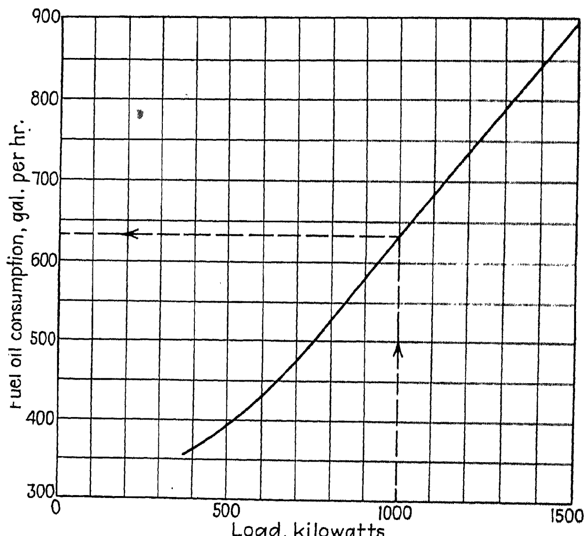

FIG. 10.—Variable load analysis using Willans-line method.

1,000 kw, the average fuel-oil consumption from Fig. 10 is 630 lb an hour, or 0.63 lb per kwhr generated. The total fuel consumed in generating 8,760,000 kwhr is the product of this quantity of electricity produced and 0.63 lb representing the quantity of oil per kilowatt-hour generated, or 5,520,000 lb annually. With a weight of fuel oil of 8.28 lb per gal, this represents 657,000 gal required annually, or an average energy production of 13.15 kwhr per gal.

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CHAPTER III

POWER-PLANT PERFORMANCE

Any study of power cost will be largely restricted to data obtained from diesel-electric units and plants owing primarily to the fact that such plants lend themselves to the exact metering of the output of the unit while the mechanical-drive plant does not lend itself readily to such measurement. There is no reason why the engineer working out a mechanical-drive problem should not accept the data obtained from diesel-electric plants. Prime-mover power is delivered as a matter of torque and rotative speed; what happens to the torque and speed beyond the flywheel is not fundamental to the prime mover itself.

12. Definition of Terms.—Definitions of terms which are employed here as well as in the annual Report on Oil-engine Power Cost prepared by the A.S.M.E. are included, as an aid in interpreting the information presented in this chapter. These reports of oil-engine-power costs referred to have been issued annually since 1929 by the Oil and Gas Power Division of the A.S.M.E. Many contend that they show the operating results of the older slow-speed engines installed in central stations and that no account has been taken in these reports of the later developments and improvements in engine design. The further objection has been voiced that these reports completely ignore the small high-speed engines which are being used more extensively year by year. While there are grounds for these objections, nevertheless, they contain the most extensive and authoritative compilation of data on diesel engines employed in central-station service in the United States.

13. Running-capacity Factor.—The average load carried when operating expressed as a percentage of the full-load rating is called the *running-capacity factor.* When it is given for an engine, it is the *running-engine-capacity factor* (RECF), and when dealing with an entire plant it is called the *running-plant-capacity factor* (RPCF). For instance, if a 300-kw unit operates for 10 hr to

generate 1,500 kwhr, the running-engine-capacity factor (RECF) for that period is 50 per cent. If a plant containing one 300-kw and one 500-kw unit generates 4,000 kwhr in a 10-hr period, during which the smaller unit operates continuously and the larger for 5 hr, the running-plant-capacity factor (RPCF) for that 10-hr period is

$$\frac{4,000 \times 100}{300 \times 10 + 500 \times 5} = 72.8 \text{ per cent}$$

In this example on running-plant capacity factor, the 72.8 per cent might or might not be typical of the operation of the separate units. For instance, the 4,000 kwhr might have been produced by the smaller unit running at 90 per cent running-engine-capacity factor to generate 2,700 kwhr and the larger at 52 per cent to generate 1,300 kwhr. It is therefore evident that any analysis of *engine* operation or cost ought to be based upon running-engine-capacity factor if obtainable, rather than on running-plant-capacity factor.

Running-capacity factors for electric generating units are usually calculated from the gross kilowatt-hour output, or the output before deductions for power-plant use of current.

14. Use Factor.—The use factor for power equipment is the amount of energy actually generated expressed as a percentage of the energy that would have been generated had the unit or plant operated at full rating continuously over the whole period, usually taken as 1 year. For example, if a 300-kw unit generated 1,200,000 kwhr in a 1-year period, the use factor would be

$$\frac{1,200,000 \times 100}{300 \times 8,760 \text{ hr}} = 45.7 \text{ per cent}$$

Use factor may be expressed by the output per unit of capacity, *i.e.*, by the kilowatt-hours produced per kilowatt capacity per annum with 8,760 kwhr per kilowatt of installed capacity per year corresponding to 100 per cent use.

Plant-use factor is usually based upon net output. It is useful to determine the unit cost for items that are independent of the amount of power generated.

15. Other Factors.—Students of the A.S.M.E. reports will find two other factors used there. These are annual-plant-load factor

and service factor. The annual-plant-load factor is the annual gross kilowatt-hour plant production expressed as a percentage of the gross energy that would have been generated had the plant operated continuously throughout the year at peak load.

Service factor is the percentage relationship between the hours operated and the total number of hours in the period. When applied to plants containing several engines, it is an average, weighted by engine size.

16. Fuel-oil Use.—Information has been assembled by the Oil and Gas Power Division of the A.S.M.E. annually since 1929 on fuel-oil consumption of diesel-engine generating units in central-station service. This information has been presented in the annual reports in the form of a series of curves showing the median fuel economy for all plants studied as well as the boundary curves for the 90 per cent of the plants exceeding the median and the 90 per cent of the plants not equal to the median fuel-economy curve. Figure 11 is a reproduction of this data presented in the 1938 Report on Oil-engine Power Cost. It will be noted from a study of this curve that fuel economy in gross kilowatt-hours produced per gallon of fuel oil is plotted against plant-running-capacity factor in per cent. It is readily apparent from this curve that as the plant-running-capacity factor increases the quantity of electrical energy produced per gallon of fuel oil consumed also increases. This is to be expected, because the best efficiency of any diesel engine is generally between three-fourths and full load on the engine.

Median curves of fuel-oil economy taken from several of the annual power-cost reports have been reproduced in Fig. 12. It will be seen from a study of these curves that there has been an improvement in the efficiency of plants, particularly in those operating above 80 per cent plant-running-capacity factor. As in the former fuel-economy curve taken from the 1938 report, the fuel economy improves as the plant-running-capacity factor increases. Curves presented in both Figs. 11 and 12 are open to the criticism that they show the operating results of engines that have been in service as long as 30 years, as well as engines that have been in service only a few years. The data, therefore, while interesting, are of little practical value when considering the performance of modern engines. That there has been an improvement in the fuel economy of diesel engines is borne out by a study

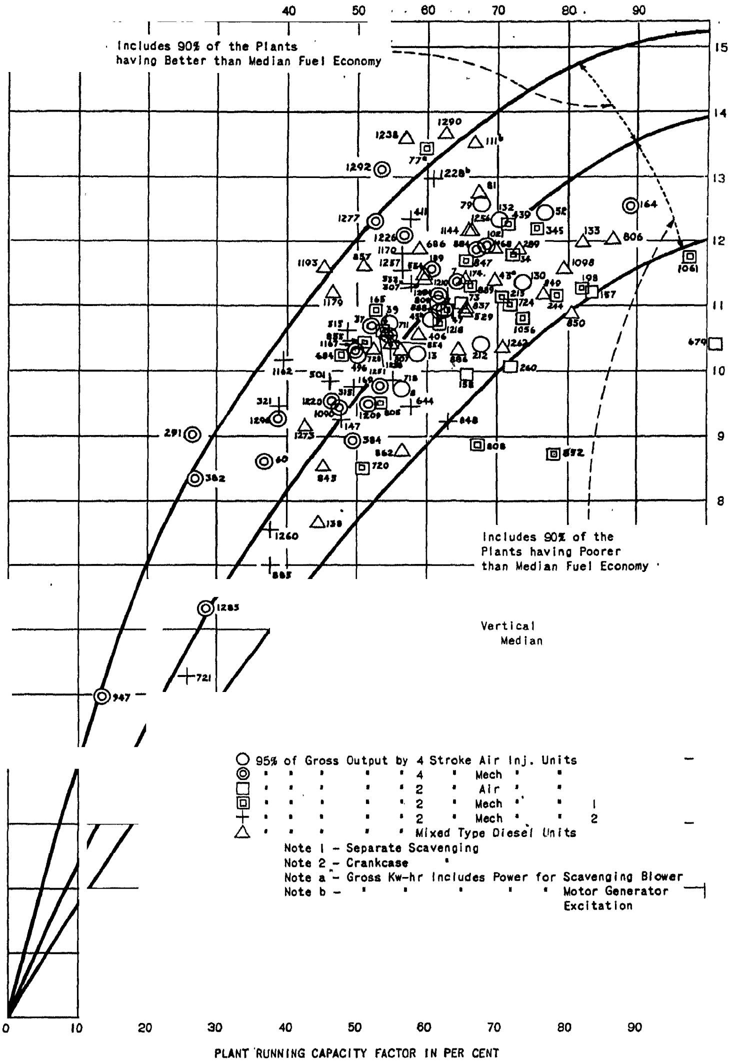

FIG. 11.—Fuel economies of 115 full-diesel plants. (*A.S.M.E. data.*)

of these oil-engine power-cost reports made by P. H. Schweitzer[1] and contained in Table 6 taken from his study.

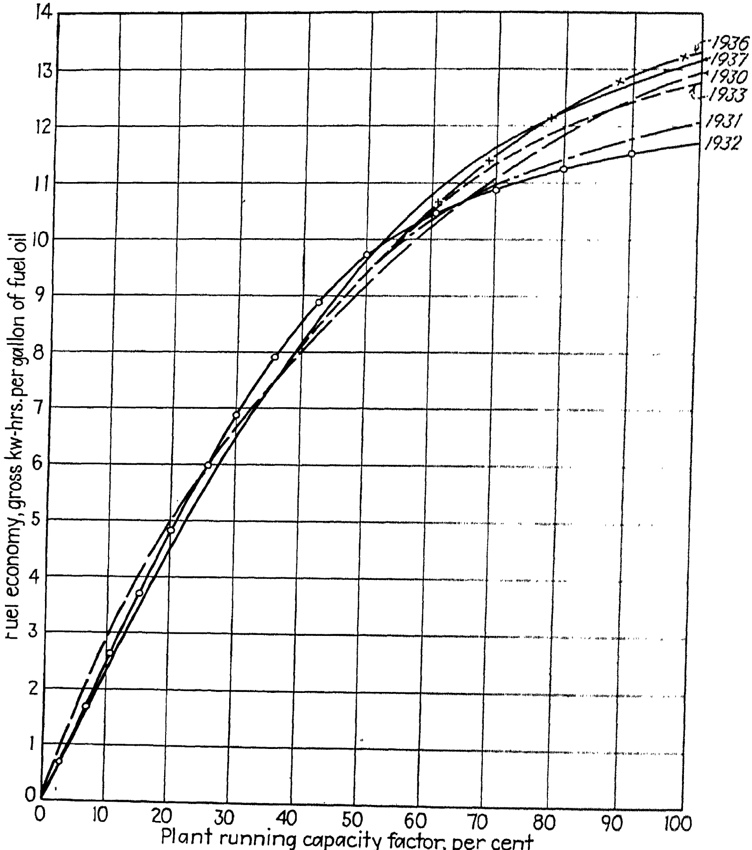

FIG. 12.—Median fuel economy of full-diesel plants for six annual reports. (*A.S.M.E. data.*)

This table was developed from a study of the median curves for the various types of engines installed between 1922 and 1937. Schweitzer goes on to state in his study:

[1] SCHWEITZER, P. H., Where Does Interpretation Begin? Analysis of the Oil-engine Power Cost Report, *Proc. O.G.P. Div. A.S.M.E.*, 1939.

The mechanical-injection engines demonstrated a phenomenal improvement in their fuel economy and took the leadership from the air-injection engines, although the latter recorded some improvements also. It is obvious that estimating the probable fuel economy of a modern engine from statistical figures which include engines of various ages would be very misleading.

TABLE 6.—EFFECT OF ENGINE AGE ON FUEL ECONOMY

Type of engine	Fuel economy, kwhr per gal		Per cent increase
	1922	1937	
Four-stroke, air-injection	10.8	12.8	18.5
Four-stroke, mechanical-injection	10.4	13.8	32.6
Two-stroke, air-injection	11.2	12.3	10.0
Two-stroke, mechanical-injection, separate-scavenging	10.7	13.4	25.2
Two-stroke, mechanical-injection, crankcase-scavenging	9.1	12.4	36.2

The question arises, is the improvement due to the better mechanical condition of the new engines or to improvements in design and construction? The answer is that the improvement is almost entirely due to better design and construction. M. J. Reed has investigated the history of 28 engines which have reported fuel economies successively for 5 years or more. Of these, three operated for less than 10,000 hr total in the longest continuous period and were therefore eliminated. The data for the remaining 25 units are shown diagrammatically in Fig. 13.

Examination of Fig. 13 does not indicate that fuel economy deteriorates progressively. Eighteen of the 25 units show better economy in the last year than the average reported. This should prove that engine age as such has no pronounced effect on the fuel economy.

This study points out the further interesting conclusions from a study of the data contained in the annual Reports on Oil-engine Power Cost:

1. The engine load has a great effect on fuel-oil economy. The greater the average load on the engine, the more kilowatt-hours are produced per gallon of fuel-oil consumed by the engine.
2. The type of engine has a small effect on fuel-oil economy.

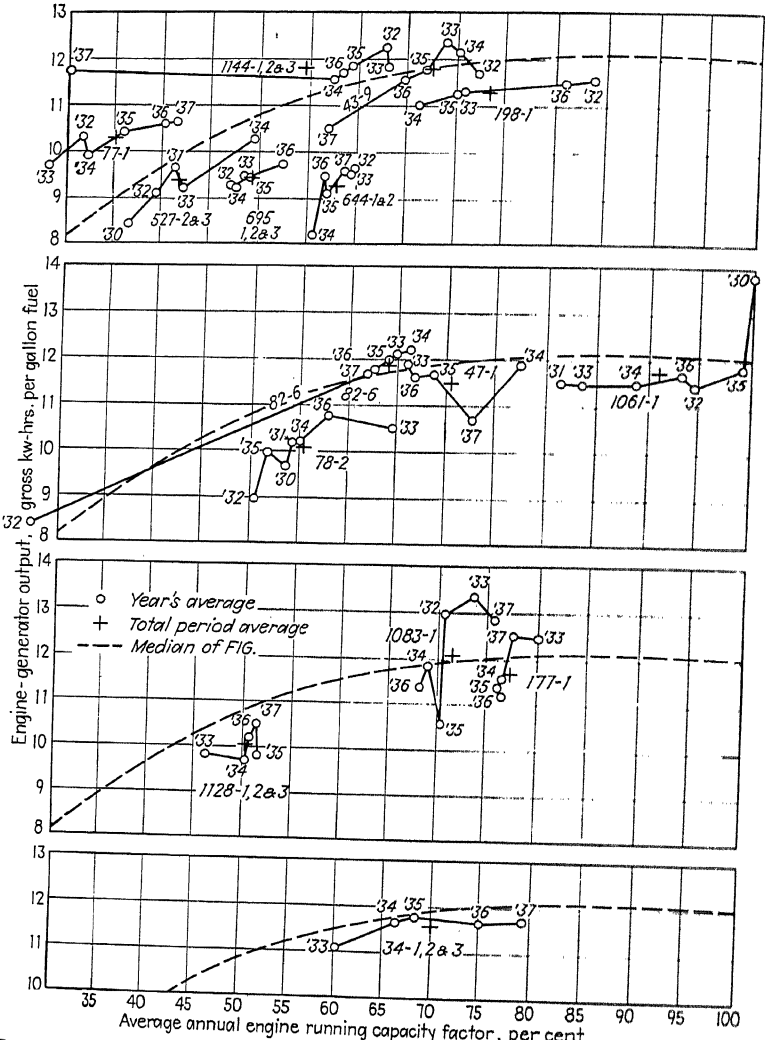

FIG. 13.—Annual average fuel economies of 25 diesel-generator sets operat… 10,000 hr or more each. (*Reed.*)

3. The year of engine manufacture has a great effect on fuel-oil economy. The more modern the engine, the better the fuel-oil economy owing to improvements in design and construction.

4. Engine size has practically no effect on fuel-oil economy.

5. Engine speed has practically no effect on fuel-oil economy.

6. Specific gravity of fuel oil has practically no effect on fuel-oil economy.

Fuel-oil-consumption guarantees for several makes, types, and sizes of modern diesel engines are contained in Fig. 14. While fuel-oil guarantees are made for only one-half, three-quarters,

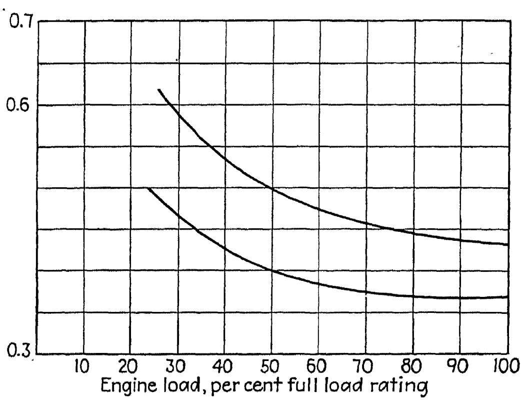

FIG. 14.—Range of fuel-oil consumption for diesel engines.

and full load, the manufacturers are requested to furnish data for fuel-oil consumption at one-quarter load on all diesel-engine projects handled by the consulting-engineering organization with which the author is associated. Figure 14 shows that diesel engines in excess of 200 hp will have relatively the same fuel-oil-consumption guarantees. Thus at full load, the variation is 0.07 lb; at three-quarters load, 0.05 lb; and at half load, 0.08 lb. Since all fuel-oil-consumption guarantees are now quoted subject to a 5 per cent tolerance, the information contained in Fig. 14 for three-quarters and full-load conditions will practically all fall within the tolerance limit.

While this curve of fuel-oil consumption can be used for preliminary studies, it is advisable before making final fuel-economy studies to obtain information from several manufacturers of diesel

engines covering the specific engine sizes required in order to obtain the latest fuel-oil-consumption guarantees. In obtaining these guarantees, care must be taken to determine what limitations, if any, are placed upon them. Usually guarantees are based upon intake air temperatures of 40 to 90 F and barometric pressure of the intake air between 28.25 and 30 in. Hg.

17. Natural-gas Consumption.—For estimating the fuel consumption of natural-gas engines, data from numerous engine manufacturers have been averaged and plotted in Fig. 15. It can be assumed for all practical purposes that the high heat value of a

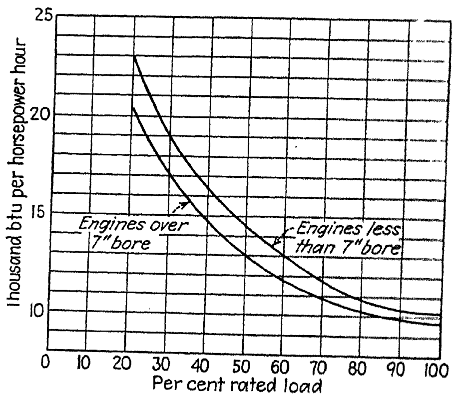

FIG. 15.—Average heat consumption of natural-gas ...

cubic foot of natural gas as purchased is 1,000 Btu. Where a more careful check of the heating value of gaseous fuels is desired, reference should be made to the data contained in Chap. X.

18. Lubricating-oil Use.—Data on lubricating-oil consumption of diesel engines installed in central-station service has been compiled annually by the Oil and Gas Power Division of the A.S.M.E. during the past 10 years. This information has been summarized in the annual reports in the form of a series of curves showing the median lubricating-oil economy as well as the boundary curves for the 90 per cent of the plants exceeding the median and the 90 per cent of the plants not equal to the median. Figure 16 is a reproduction of these data presented in the 1938 Report on Oil-engine Power Cost covering that year only. It will be noted that

the curves given in this figure are all straight lines. They have been constructed as straight lines on the theory that lubricating-

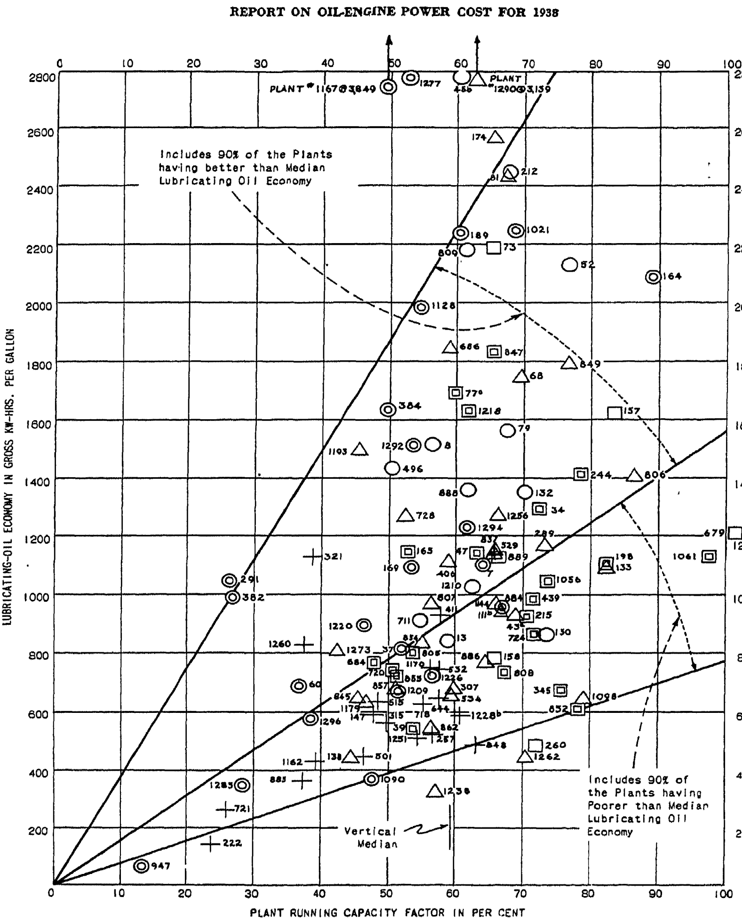

FIG. 16.—Lubricating-oil economies of 114 full-diesel plants. (*A.S.M.E data.*)

oil consumption of any engine is a constant at constant speed regardless of load. This may not be strictly true, but experience

dictates that it is sufficiently accurate for all practical purposes. Thus, as the plant-running-capacity factor increases, the kilowatt-hours generated for each gallon of lubricating oil consumed also increase.

The various engine builders do not make guarantees on lubricating-oil consumption because of the fact that operating personnel and plant conditions have such a great effect on lubricating-oil consumption. Since guarantees are not made on lubricating-oil consumption, and since lubricating-oil consumption cannot be accurately determined except on test runs of long duration or from actual operating records, the only accurate information on lubri-

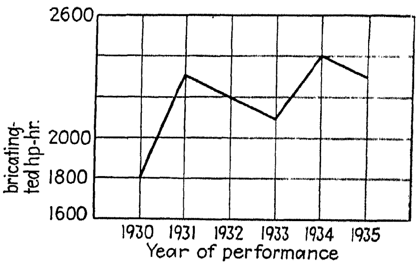

FIG. 17.—Lubrication performance of the same 74 diesel for the years 1930–1935. (*Reed.*)

cating-oil economy is contained in the annual Reports on Oil-engine Power Cost.

Plant operators very often tend to overlubricate their engines and particularly to overlubricate cylinders. The effect of such excessive lubrication is twofold. Lubricating oil, like fuel oil, will burn, and excess lubrication will tend to show a decrease in fuel-oil consumption. Lubricating oil burned in the cylinder leaves behind carbon which may be blown out the exhaust, or remain to cause stuck rings and valves.

For estimating lubricating-oil consumption in connection with economic studies, one can usually assume 2,000 to 3,000 rated horsepower hours per gallon of lubricating oil. Thus a 100-hp engine would consume a gallon of lubricating oil every 20 to 30 hr of operation.

M. J. Reed has investigated the lubricating-oil economy of 74 identical engines over a period of 6 years and obtained Fig. 17, which shows an insignificant variation in lubricating-oil economy

as the engine gets older. The lubricating-oil economy of an engine once installed does not remain constant but usually varies in cycles. In the second year the lubricating-oil economy is, as a rule, better than in the first year because the engine has run in. Then for 4 or 5 years, the economy slightly deteriorates because of cylinder and ring wear. After 5 or 6 years, there is a general overhaul, involving replacement of piston rings, and sometimes liners. After that the lubricating-oil economy is about equal to that of the new engine and the cycle repeats. *Therefore the engine age as such can be ignored as a factor in lubricating-oil economy.*

Schweitzer arrived at the following conclusions based upon his study of the annual Reports on Oil-engine Power Cost.

1. The engine load has a great effect on lubricating-oil economy. The greater the average load on the engine, the more kilowatt-hours are produced per gallon of lubricating oil consumed.

2. The type of engine has a great effect on lubricating-oil economy.

3. The year of engine manufacture has a great effect on lubricating-oil economy. The more modern the engine, the better the lubricating-oil economy, owing to improvements in design and construction.

4. Engine size has practically no effect on lubricating-oil economy.

5. Engine speed has practically no effect on lubricating-oil economy.

19. Maintenance.—The maintenance of diesel engines, as pointed out by Spencer in a paper before the Oil and Gas Power Division of the A.S.M.E., ranges from the fix-it-when-it-breaks idea to a very finely worked out system of inspections so complex as to be of questionable economic value. There is no argument against the fact that adequate and conscientious maintenance of equipment results in longer life for the equipment and better operating performance.

In this connection it is interesting to consider some of the complaints that have been raised by plant operators in regard to difficulties they have experienced with equipment under their supervision. The plant operator often loses sight of the fact that equipment does wear out and that mechanical difficulties will confront him from time to time. Many operating men fail to

realize that if no part ever wore out or no failure of such a part occurred it would be possible to lock up the doors of the plant, discharge the operating staff, and merely have an attendant come and inspect the equipment in operation once a week, primarily to ensure that sufficient fuel and lubricating oil were available for another week's operation. The subject of reliability of diesel-engine equipment will be discussed elsewhere in this book. It is intended only to point out here that equipment must be watched constantly and that the reward of constant vigilance is fewer mechanical difficulties and lowered maintenance costs.

While maintenance is a thing which starts when an engine or any other item of mechanical or electrical equipment is put into operation, it is advisable to know what we may expect in the way of maintenance costs for equipment we are contemplating installing. The best source of information pertaining to maintenance costs in diesel power plants, and particularly for those plants utilizing slow-and medium-speed engines, is the Report on Oil-engine Power Costs to which previous reference has been made. These reports afford the basis for the following cost information on maintenance.

From the 1937 cost report, a total of 47 plants with engine-maintenance records of five or more years with the same installed engine horsepower were analyzed. From this analysis it was found that the engine-repair costs for 65 per cent of the plants was $1 or less per installed brake horsepower per year, while the remaining 35 per cent had engine-repair costs in excess of this amount. This study further indicated that regardless of whether or not the plant was operated continuously or used for peak-load and stand-by service the maintenance costs were practically the same for both types of plants. Likewise, the maintenance cost apparently was not influenced much by engine horsepower per unit or the number of units installed.

The same report gave 70 plants for which the record of the total supplies, repairs, and miscellaneous plant costs was available for periods of 5 or more years on the same installed horsepower through the period studied. According to the definition of terms in the questionnaire sent out by the Oil and Gas Power Division of the A.S.M.E. for compiling the Report on Oil-engine Power Cost, *supplies* are defined as "items used in the power generating plant which are consumed in the operating process, the replacement of

which does not constitute a repair or renewal." The principal items of supplies are listed as "waste, packing, wipers, gauge glasses, gaskets, bolts, screws, nails, dynamo and motor brushes, cans for containing rags and waste, transformer oil, or hand oil cans." *Miscellaneous* is defined as being items such as "lighting, heating, and cleaning systems, fire-protection systems, janitor supplies, ice water, meals and carfare, stationery, telephone and

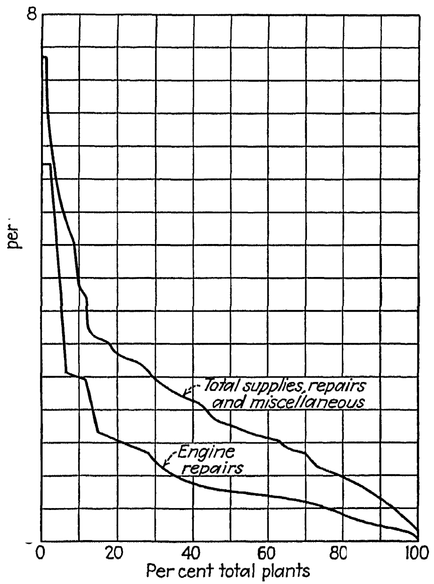

FIG. 18.—Engine repair and total supply, repair and miscellaneous costs for diesel-electric plants. (*A.S.M.E. data.*)

toilet service, care of streets, yards and sidings." While this cost in terms of dollars per brake horsepower per year is greater than engine-repair costs owing to the inclusion of other items listed above, these figures likewise indicate that it apparently does not make much difference whether the plant is operated continuously or for peak-load service insofar as the cost for supplies, repairs, and miscellaneous items is concerned. In this instance, the installed horsepower does appear to have some bearing upon the total cost of supplies, repair, and miscellaneous costs.

From this study, Fig. 18 was developed to show both engine-repair costs and total supplies, repairs, and miscellaneous costs in a form which indicates the percentage of the plants investigated in which the cost per brake horsepower per year exceeded a given value in dollars. For example, referring to the two curves in Fig. 18, it is seen that engine-repair costs exceeded $1 per brake horsepower per year in 35 per cent of the plants studied, and that the total for supplies, repairs, and miscellaneous costs exceeded $2.25 per brake horsepower per year for 35 per cent of the plants.

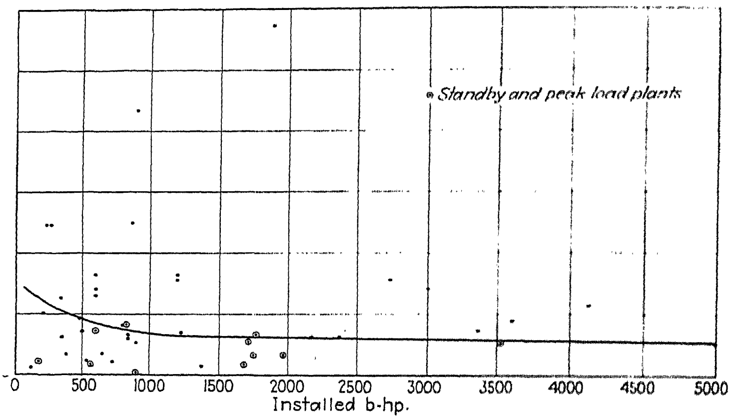

FIG. 19.—Engine repair costs. (*A.S.M.E. data.*)

Likewise 50 per cent of the plants had engine-repair costs less than 80 cents and total supply, repair, and miscellaneous costs of less than $1.75.

The data pertaining to engine-repair costs are shown in Fig. 19 in relation to installed horsepower in the plant. In Fig. 20 is presented information showing the total supply, repair, and miscellaneous costs in relation to the installed horsepower in the plant.

The apparent lack of uniformity of pattern to the data in Figs. 19 and 20 can be explained only by assuming that those plants which show the highest maintenance costs are operated on the theory "let it break first." Experience with diesel plants and diesel-plant operators over a period of years leads one to the

conclusion that *the operator who does an adequate maintenance job on a periodical schedule which is never varied has nothing to worry him.* As one very successful diesel-plant superintendent puts it, "If I neglect my maintenance I'm playing on borrowed time."

As proof of the ability of a plant superintendent or chief engineer to reduce maintenance costs and improve reliability of diesel prime movers in electric-generation duty, the following maintenance record covering 9 years of operation of the municipal power plant at Ponca City, Okla., is given.

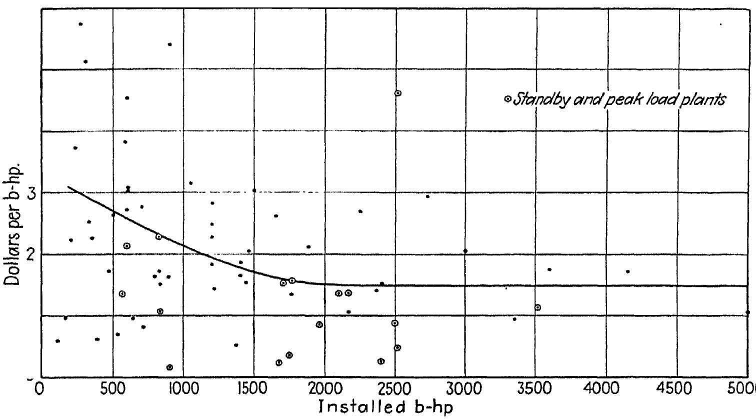

FIG. 20.—Total supply, repair, and miscellaneous costs. (*A.S.M.E. data.*)

This record, covering primarily the four engines installed from 1924 to 1928, is all the more impressive in light of the fact that the fuel used during this period showed a specific gravity ranging from 10 to 11° Bé; flash point, 265 F; carbon residue (Conradson), 9 to 10 per cent; viscosity at 100 F (Saybolt universal), 452; and asphalt at 100 F penetration, 43.9 per cent. During the year 1936 the four air-injection units then installed averaged 10.1 kwhr per gal of fuel oil and 2,249 kwhr per gal of lubricating oil.

20. Plant Labor.—In order to determine the requirements for plant labor, reference has been made to the information contained in the Report on Oil-engine Power Costs for the year 1937. This report contained 75 plants for which complete labor information was available. Peak-load and stand-by plants were elimi-

TABLE 7.—MAINTENANCE DATA, PONCA CITY, OKLA.

Engine	Units installed	Date installed
1	1,250 hp Nordberg type EG, 2-cycle, air injection	January, 1924
2	1,250 hp Nordberg type EG, 2-cycle, air injection	March, 1926
3 and 4	1,250 hp Nordberg type EG, 2-cycle, air injection	April, 1928
	2,250 hp Nordberg type TA-216, 2-cycle, air injection	November, 1937

Year ending June 30	Kwhr produced	Maintenance cost		
		Total	Per horsepower installed	Per kwhr
1929	7,955,600	$ 3,873.55	$0.774	$.00487
1930	8,470,800	2,671.00	0.534	.00315
1931	8,839,200	2,660.85	0.531	.00301
1932	8,718,700	1,621.90	0.325	.00186
1933	8,624,600	2,373.38	0.475	.00275
1934	9,161,800	3,580.18	0.717	.00392
1935	9,637,400	3,205.19	0.640	.00333
1936	10,115,500	3,849.04	0.770	.00381
1937	11,017,400	4,538.74	0.908	.00413
Total...........	82,541,000	$28,373.83		
Average per year...	9,171,222	$ 3,152.65	$0.630	$.00344

Year ending Dec. 31	Engine No. 1	Engine No. 2	Engine No. 3	Engine No. 4
	Hours operated			
1929	5,103	5,728	2,971	913
1930	5,332	6,155	2,606	1,473
1931	5,572	6,345	3,167	736
1932	5,261	5,614	2,444	1,938
1933	5,181	5,765	2,247	2,616
1934	3,600	4,683	5,104	3,416
1935	3,494	5,608	5,371	2,809
1936	5,083	6,042	4,568	2,937

Major parts replaced in addition to one set of cylinder liners for each engine

Engine No. 1	Engine No. 2	Engine No. 3	Engine No. 4
2 bottom halves main-bearing shells 1 crosshead bearing 1 rebabbit crank-pin bearing 1 inner cylinder head	1 inner cylinder head 1 rebabbit crank-pin bearing 3 crosshead bearings	3 inner cylinder heads 3 rebabbit crank-pin bearings 1 piston head 4 crosshead bearings 1 liner for high-pressure compressor	1 set camshaft gears 1 crosshead bearing 1 rebabbit crank-pin bearing 1 inner cylinder head

Balance consists of piston rings, compressor valves, miscellaneous small parts, and gaskets. These engines are still running with original piston heads, piston skirts, and wrist pins except for one piston head replaced in No. 3 unit.

nated from consideration since the labor for such plants is subject to too many factors outside of equipment operation. When consideration is being given to a plant for peak-load or stand-by power service, labor requirements should be given special consideration.

These data indicated that when the installed capacity is 1,000 hp or less it is practically universal to use one man per shift, and when the plant exceeds this installed horsepower, the

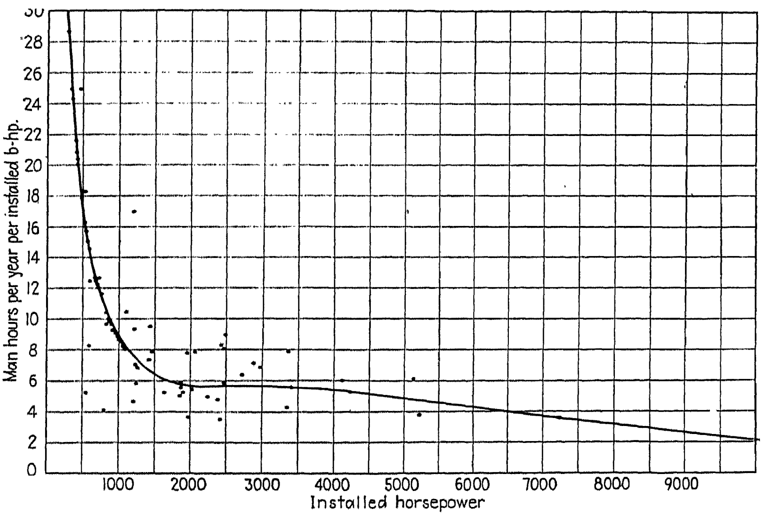

FIG. 21.—Plant labor, man-hours per year per installed brake horsepower. (*A.S.M.E. data.*)

number of men per shift increases. This information is given in Fig. 21 and is shown as man-hours per year per installed brake horsepower for various sizes of plants. For example, with a plant having an installed capacity of 2,000 hp, it appears that customary practice would call for approximately 5.5 man-hours per installed brake horsepower per year, or a total of 11,000 man-hours of plant labor for the year. On the basis of three 8-hr shifts per day, this represents 1.25 men per shift on the average. This could be interpreted to mean that three shifts are operated with one man per shift, with a fourth man called upon for part-time and relief operation.

The whole subject of the number of operators per shift in any plant is one that calls for considerable study of the plant in question. Not only is the number of operators per shift influenced by the size of the plant as determined by the horsepower capacity, but the number of engines, number of engine cylinders, design of plant, and the possibility of operating the engine equipment in conjunction with other facilities as in a water-works plant, for example, may influence the size of the plant staff.

A study of wages paid to operators in diesel-engine plants and as reported by the A.S.M.E. indicates that operators can be secured at rates corresponding roughly to the wages paid for comparable work in comparable trades. Wage rates for operators vary over the country and are influenced largely by the prevailing rates for skilled craftsmen in the particular locality. A study made from the 1935 Report on Oil-engine Power Cost indicated that the lowest rate reported was 18 cents per hour, the highest $1.52, and the median 52 cents per hour. This study was based on data supplied by 103 plants scattered throughout the United States.

In determining the labor cost for any particular plant, it is necessary to determine the number of men required per shift, and the probable wage rates to be paid in the locality where the plant is being considered. These two factors will permit determination of the labor cost for plant operation.

21. Auxiliary Power Requirements.—In a plant of any size, it is rare that all the power delivered at the generator terminals is available to handle the ultimate load. Usually some is required to operate cooling-water pumps, lubricating-oil centrifuge motors, starting-air compressor, fuel-oil-transfer pumps, for lights, and to make up for electrical losses between the generators and outgoing switchboard panels. Very small diesel units may mount all equipment as engine attachments, driven by gears or chains from the crank or cam shaft. In such cases, there is virtually nothing to subtract from the generator output because the needed power has already been subtracted before the main output is delivered to the engine flywheel. For very large engines, more equipment may be motor driven as, for example, the scavenging blower.

There is a wide discrepancy in the percentage of gross power that is used in the power plant. One situation responsible for

high percentages is the practice, undesirable though it is, of using electric power for the heating of lubricating oil before centrifuging. On the other hand, plants pumping city water often divert part of it from the outgoing main for engine cooling and thereby show a fictitiously low use owing to the absence of power generally necessary for pumping cooling water.

In Fig. 22 are shown the percentages of gross output for station use plotted against running-plant-capacity factor for 131 plants (all giving such data) listed in the power-cost report for

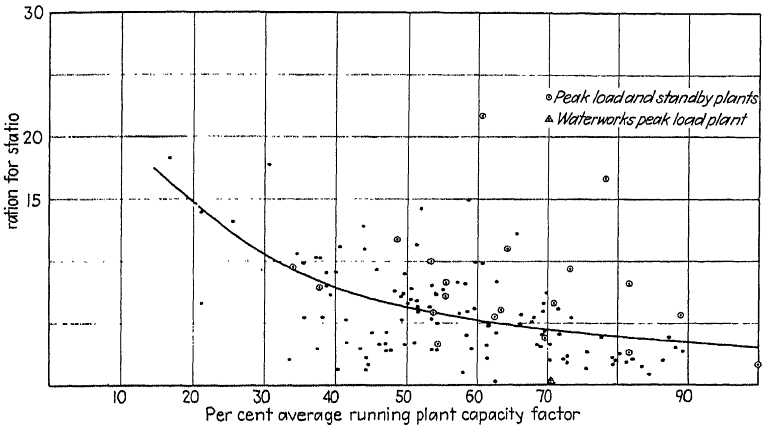

FIG. 22.—Electricity required for station auxiliaries and lighting in per cent of gross generation. (*A.S.M.E. data*, 1937.)

the year 1937. The wide swing in values is quite apparent. However, order can be arrived at by adopting a theory that station requirements are constant regardless of running-plant-capacity factor. As a matter of fact, this theory is about correct. The cooling-water pumps are rarely slowed down or throttled for reduced load, the air compressor is operated for about the same period, the lubricating-oil centrifuge goes along at the same rate, the lights are needed just the same, and the losses are probably nearly constant. The curve in Fig. 22 is drawn so that half the points are above, half below. It is also constructed so that the percentages result in constant energy for station use. You will note, for example, that 3.15 per cent of full load is equal to 6.3 per cent of half load.

If the assumption can be made, as seems likely, that plants using electric heaters or other wasteful equipment balance other plants by-passing pay-load water for cooling, this median is a fair indication of what the power-plant use *should* be. The average engine of the report is one that is self-contained as to fuel injection, scavenging, and lubricating-oil pumping, but not so in respect to cooling-water, centrifuge, starting-air, fuel-transfer, and lighting load. Therefore, the curve of Fig. 22 applies to such units. Corrections should be made when applying this curve to any other auxiliary arrangement.

This curve illustrates the relationship between gross and net demand or peak. For example, if the net peak load is to be 500 kw, the gross peak will be 500/0.965 or 518 kw, assuming that it is planned to handle the peak at 90 per cent of rating, where the use of gross is shown to be 3.5 per cent.

CHAPTER IV

ECONOMIC STUDIES

So far this book has been devoted to a discussion of variable load conditions imposed upon electric generating stations and the factors affecting the operating costs of equipment. In this chapter, in furtherance of the preliminary study of the adaptability of the internal-combustion engine for prime-mover duty, matters of equipment and building costs will be considered, as well as a study of the selection of units and their reliability.

22. Cost of Diesel Plants.—Any cost information presented here must be very general. It is obviously impossible to present data that will cover all points involved in making cost estimates or to anticipate changes in price levels. It is necessary, therefore, to check equipment prices thoroughly whenever a close estimate of construction cost is necessary by obtaining current quotations from the equipment manufacturer.

In Fig. 23 engine and generator prices current in January, 1935, are shown. The engine prices covered within the range of the two upper curves shown in this figure include the engine, flywheel, pyrometer, exhaust muffler, intake-air muffler, air filters, starting-air plant, fuel-oil day tank, cooling-water pumps, oil cooler if required, temperature alarms, and all piping for the engine together with the necessary erection labor for installation. At the bottom of this figure generator and exciter prices have been included, using 2,400-volt three-phase 60-cycle generators and direct-connected 125-volt exciters. These prices have been reduced to unit price per engine brake horsepower so as to make it possible to add the generator cost per horsepower of engine capacity directly to the engine price to obtain the cost of the entire unit per engine brake horsepower.

No addition has been made in Fig. 23 for lubricating- and fuel-oil conditioning equipment, exhaust heat boilers, switchboard and wiring, cooling system, oil-storage and handling equipment, engine foundations, or building necessary for housing the equipment.

While the prices contained in Fig. 23 were current in 1935, they have been fairly constant during the past several years. In all probability they are somewhat low at the present time (January, 1942) and should be checked when an actual installation is contemplated. They do furnish, however, a ready check of the

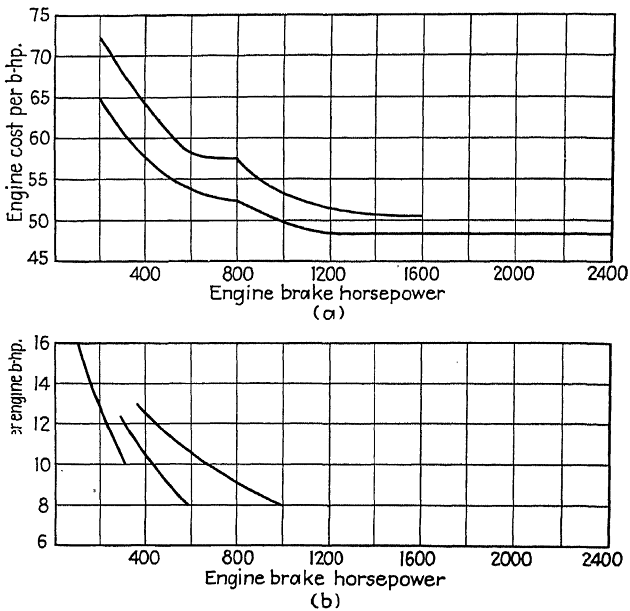

FIG. 23.—Cost of diesel engines and generators, dollars per engine brake horsepower, 1935–1938.

relative costs of engine-driven generating units of the various sizes covered by the chart.

From an examination of this figure it is seen that a relatively wide range of prices per engine horsepower exists in the smaller horsepower sizes. This is due, primarily, to the difference in the rotative speeds of the engines considered. While, in general for a given horsepower rating of an engine, the higher the rotative speed, the lower the cost of the engine per brake horsepower, this is not always true. Therefore, it cannot be considered that the

upper curve represents the slower rotative speed while the lower curve represents the higher rotative speed engines.

An additional guide in checking estimates of diesel-electric generating stations is contained in Fig. 24. The lower curve in this chart shows the cost of all plant equipment including two diesel engines, two generators, two exciters, mufflers, air filters, switchboard and wiring, cooling equipment, fuel-oil storage tanks, station piping, and all other equipment and erection costs exclusive of the building and foundations required. Both generating units included are of equal size. The upper curve

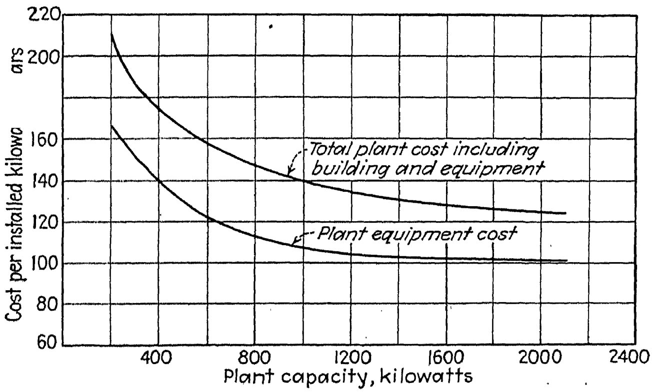

FIG 24.—Cost of complete diesel-electric generating stations based upon 1938 data.

gives the cost per kilowatt of the entire plant including the equipment as well as the building and foundations. Data from which Fig. 24 was developed contained price information current during 1939 and 1940, although existing (January, 1942) price conditions make this material somewhat low for present market conditions. This pair of curves does show, however, that after the plant capacity reaches approximately 1,400 kw the cost per installed kilowatt of generating capacity remains practically constant up to 2,150 kw.

These data have been verified during the past several years by the author's own experience. However, it is advisable at this point to offer a word to those who read these pages on the matter of cost information. *No cost information is of any value which*

not been accumulated by the person who uses it. Anyone assembling cost information to be used for developing estimates of a particular installation should consult with equipment manufacturers as to the prevailing prices of equipment and as to the probable future trend of these costs. Under the conditions existing today, cost information will be good only during the immediate future, and it may be higher or even lower tomorrow. When it is realized that the equipment in a diesel- or other internal-combustion-engine power plant represents 85 to 90 per cent of the total plant cost, it is readily apparent why it is necessary to obtain from the equipment manufacturer estimating data with which to prepare an up-to-the-minute cost of the project contemplated.

23. Building Estimating Data.—In addition to the cost of the diesel-engine generating or power units in a plant, the cost of building construction, power wiring, switchboard costs, and other accessory equipment not included in the engine manufacturer's figures must be included to complete the necessary estimate of cost.

The organization with which the author is associated has been assembling data on costs of power-plant buildings constructed during the past 10-year period. The data obtained from a representative group of these buildings are contained in Table 8 which shows the date of contract, building volume, structural steel per cubic foot of building volume, and the cost per cubic foot of building volume. All structures were equipped with reinforced-concrete substructure and structural-steel and brick superstructure.

Building costs should be estimated high, probably considerably higher than the values given in the accompanying table. The reasons for "raising the sights" on preliminary estimates are twofold. In the first place, the construction of the project may be delayed one or two years after the preliminary estimates have been prepared, introducing unforeseen contingencies and possible increases in cost. In the second place, information is not always available when the preliminary estimate is made which will permit the estimator to make a close check on building-construction costs.

24. Switchboards and Wiring.—Other items of cost such as switchboards and wiring should be estimated liberally. For

TABLE 8.—CONTRACT COSTS OF DIESEL-POWER-PLANT BUILDINGS

Date	Capacity		Building volume, cu ft	Weight structural steel per cu ft, pound	Cost, cents per cu ft		Notes
	Hp	Kw			Building	Building and foundation	
1934	2—375	2—250	59,400	0.70	....	22.0	
1934	1—375 1—450	1—250 1—300	57,300	0.64	28.1	32.5	PWA job
1935	2—450	1—300	59,400	0.70	33.5	38.7	
1936	3—650	3—447	147,400	0.58	....	34.6	PWA job. Space provided for fourth unit
1938	2—600	2—400	57,300	0.64	28.3	32.8	PWA job
1939	2—800 1—600	2—560 1—450	164,700	0.79	....	31.6	PWA job
1940	3—1,000	3—700	177,000[a]	0.44	....	34.5	PWA job
1940	1—1,500 2—1,000 1—600	1—1,100 2—700 1—400	303,000[a]	0.63	....	26.8	Space provided for fifth unit

NOTE.—Building volumes computed on measurements from center lines of columns and from roof deck to bottom of basement slab.

[a] Building volume computed on out-to-out of walls and from roof deck to bottom of basement slab.

TABLE 9.—DIVISION OF BUILDING COSTS FOR THREE RECENT PROJECTS

Item	1936 (Iowa)	1940 (Iowa)	1940 (New York)
Excavation, backfill, and disposal	2.0	2.4	4.6
Concrete, formwork, and reinforcing steel	33.0	26.2	29.5
Walls, including brick, wall tiles, stone, windows, doors, and plastering	36.0	24.7	25.8
Structural steel and miscellaneous iron	17.7	14.8	11.2
Roof, including roof slabs, insulation, roofing, cants, and flashing	6.1	4.3	4.6
	94.8	72.4	75.7
Heating, plumbing, air conditioning, and lighting	5.2	6.7	7.7
Miscellaneous equipment	0	20.9	16.6
	100.0	100.0	100.0

example, it was a matter of common knowledge that with open-type switchboards having oil circuit breakers of modest interrupting capacity the cost could be taken at $1,000 per panel, which figure included the cost of the necessary switchboard panel, instruments, oil circuit breaker, necessary power and control wiring to the generating unit or power circuit, and the installation labor. Today this figure for small plants would probably be nearer $1,500 per circuit. With the use of metal-clad switchgear, and circuit breakers of higher interrupting ratings in anticipation of increased plant capacity in the future, the cost per circuit would be even higher, ranging from $2,000 to $3,000 per circuit.

25. Piping costs for internal-combustion-engine power plants will vary considerably, owing partly to the type of cooling system employed and partly to the desire of the designer or owner to install special accessory equipment or unusual piping arrangements. The cost of the piping system for a closed or double-circuit cooling system is always more expensive than that for an open or single-circuit system since two independent circulating systems must be installed.

An examination of the piping in many diesel plants clearly shows that no two plants are provided with the same piping arrangement. Furthermore, this examination has shown that very few piping systems for diesel plants have been designed before the mechanical work of pipe cutting and fitting was started in the field, with the natural result that too often the piping cost was excessive and the operating was anything but simple. In view of this situation, therefore, it is not surprising to find the piping costs in two plants having the same number of engines of equal horsepower varying by as much as 50 to 80 per cent.

From data available on a number of internal-combustion-engine power plants, it appears that when an open or single-circuit cooling system is utilized and no piston cooling is required the piping cost will range from $1.50 to $2 per installed brake horsepower. When a closed cooling system is required, and where piston cooling is necessary, the piping cost will range from $2 to $2.50 per installed brake horsepower. In general it has been found that piping costs consist roughly of 50 per cent material cost and 50 per cent labor and overhead costs. The largest

portion of the cost of piping in an internal-combustion-engine plant having over 500 hp installed capacity is in the cooling-water system. This condition results from the necessity of employing cooling towers or spray ponds in a majority of cases for heat dissipation where relatively large quantities of water must be transported a considerable distance between the engines and the cooling tower or pond.

26. Cost of Traveling Cranes.—Relative costs of traveling cranes for use in maintenance of large engines and generators are given in Table 10. A 5-ton hand-operated crane having a 40-ft span will cost between $1,000 and $1,200 at the factory.

TABLE 10.—RELATIVE COSTS OF CRANES (40-FT SPAN)

Capacity, tons	Hand operated	Motorized hoist only	Three-motor crane
5	100	140	280
10	130	190	360
15	200	280	480
20	235	335	565
25	245	365	625
30	280	420	710

27. Fuel-oil Prices.—The cost of fuel oil is influenced by the same economic laws as other commodities. Prices rise as the demand increases and fall when buying falls off. Variations in prices of 32–36 A.P.I. gravity fuel oil as quoted by Oklahoma refineries are given in Fig. 25 and include average yearly prices as well as the variation in price at the refinery from 1925 to 1941. This grade of fuel is somewhat higher than would be purchased for a diesel power plant and is given to illustrate the price variation over the past 17 years. Variations in price per barrel of heavy residual fuel oils at Oklahoma refineries are shown in Fig. 26. The *Oil and Gas Journal* carries current fuel-oil quotations at the major refining centers each week.

Studies of variations in fuel-oil prices show that they tend to follow coal prices. This probably is due to the fact that many steam plants, equipped to burn either coal or oil, buy that fuel which is the cheaper. As a consequence, this shifting from one fuel to another depending upon market conditions tends to make oil prices rise as coal prices increase and fall as coal prices drop.

28. Reliability of internal-combustion engines, and diesel engines particularly, has been a subject of study and investigation

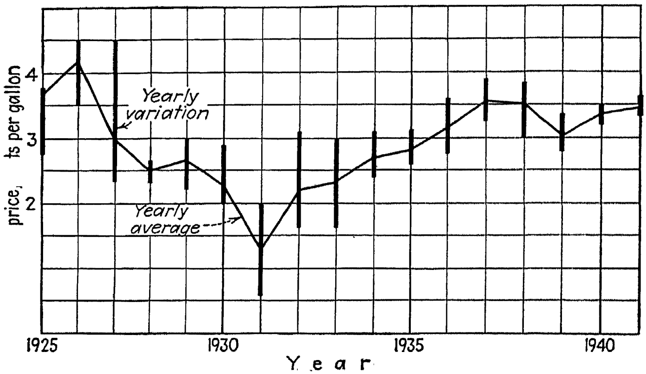

FIG 25.—Variation in price of 32-36 A.P.I. gravity fuel oil, fob Group 3 refineries, Oklahoma.

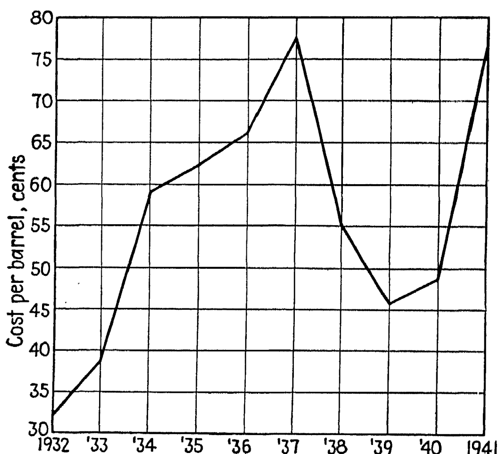

FIG. 26.—Variation in price of 8-14 A.P.I. gravity fuel oil, fob Group 3 refineries, Oklahoma.

over a period of more than 10 years. Since 1929, the annual Reports on Oil-engine Power Cost prepared by the A.S.M.E.

have enumerated data regarding enforced shutdowns of engines in operation as well as the hours each engine was out of service owing to such shutdowns.

In defining an enforced shutdown, the 1938 questionnaire sent out by the Oil and Gas Power Division of the A.S.M.E. states that "An enforced shutdown is a stoppage because of actual or imminent engine trouble. The duration of an enforced shutdown is the time *necessary* to correct the trouble which caused the shutdown. A prearranged shutdown for maintenance work is not considered an enforced shutdown." In this definition of enforced shutdown failure of apparatus other than the engine is not considered, nor is time down for regular routine maintenance of the engine included.

An analysis of enforced shutdowns as reported in these power-cost reports has been made for the decade 1930 to 1939. During the period considered in this analysis a total of 2,642 engines were reported with complete information regarding enforced shutdowns and the time necessary for making repairs. Each of these engines was reported for an entire year, so in reality this analysis deals with 2,642 engine years. The total hours of operation of these engines were 8,839,209, or an average of 3,540 hr of operation per engine per year. During this period considered there occurred a total of 1,955 enforced shutdowns, or an average of 0.74 enforced shutdown per engine per year. Of the total of 2,642 engine years reported, over 75 per cent reported no enforced shutdown time.

The complete analysis of the enforced shutdowns based upon the hours of operation per year of the engines is set forth in Table 11. In preparing this tabulation, it was considered that while no enforced shutdowns might be reported for some engines, nevertheless these engines were grouped with those having less than 1 per cent enforced shutdown time. This table shows that the total hours of enforced shutdown averaged 1.08 per cent of the total operating hours and that the highest percentage of time down was with engines operating less than 1,000 hr per year. This table further shows that enforced shutdowns in reality have no relation to the number of hours an engine is operated annually.

Any machine made by human hands is subject to failure; it is therefore impossible to ensure that no failures will occur. The data contained in Table 11 indicate that the chances of the

enforced shutdown time exceeding 1 per cent of the operating time is 1 in 10. In fact, the chances are three out of four that there will be no enforced shutdowns at all. By dividing the probability of an enforced shutdown into 20 equal parts, there are 15 chances that no enforced shutdowns will occur, 3 chances that there will be one or more with enforced shutdown time not exceeding 1 per cent of the operating time, and 2 chances that enforced shutdown time will exceed 1 per cent of the operating time. The probability that enforced shutdowns will occur on two units simultaneously in a given plant is very small.

TABLE 11.—RELIABILITY OF DIESEL ENGINES

Hr per engine	Number of engines	Total operating hr	Enforced shutdowns					Engines with shutdown time			
			Number	Total hr	Per cent operating hr	Average number per engine	Average hr per engine	Less than 1 %	1–2 %	2–5 %	Over 5 %
0–1,000	537	230,138	127	14,845.3	6.45	0.24	27.65	490	6	12	29
1,001–2,000	320	466,922	146	23,949.9	5.12	0.46	74.90	279	9	9	23
2,001–3,000	441	1,116,619	235	16,038.5	1.44	0.53	36.25	398	12	13	18
3,001–4,000	420	1,970,672	265	10,401.2	0.53	0.63	24.78	380	13	12	15
4,001–5,000	430	1,938,811	321	12,108.4	0.63	0.75	28.15	399	7	16	8
5,001–6,000	239	1,312,996	247	9,781.5	0.75	1.03	41.00	210	10	9	10
6,001–7,000	132	846,364	251	3,801.8	0.45	1.91	28.80	118	8	5	1
7,001–8,000	82	613,024	318	4,727.8	0.77	3.88	57.60	69	2	7	4
8,001+	41	343,663	45	177.5	0.05	1.10	4.33	41	0	0	0
	2,642	8,839,209	1,955	95,831.9	1.08	0.74	36.20	2,384	67	83	108

Studies have also been made as to the effect of engine age upon enforced shutdowns. Apparently age has no appreciable effect upon the number of enforced shutdowns, and there appears to be no reason to anticipate that as an engine becomes older the number of enforced shutdowns will increase annually.

29. Economic Studies.—In Chap. I, attention was called to the fact that whenever power is required, it is necessary to determine the most economical source of that power. The information presented in the chapters following, as well as that contained in this chapter, should be considered in the preliminary studies of power problems involving consideration of internal-combustion

engines. The application of this information in actual cases will now be discussed.

The cost of power produced by mechanical and electrical equipment is usually considered to consist of (1) *operating costs*, which include fuel, labor, taxes and insurance, equipment maintenance, and miscellaneous supplies, and (2) *investment costs*, which are the costs of machinery and equipment, made up of interest and principal payments on the capital invested in the power-producing machinery. Extensive discussions of the various means for developing these costs are contained in the literature on engineering economics. Several examples are given to show how these economic computations are made.

Example I. *Power for Driving Pump.* A farmer wants to irrigate a tract of land for a period of 5 months a year. It requires 50 hp to drive the pump, and either an internal-combustion engine or an electric motor drive can be used. The engine installed and ready to operate will cost him $3,000. Fuel oil costs 6 cents per gallon delivered and weighs 7.5 lb per gal. Lubricating oil costs 50 cents per gallon. An electric motor and control for the installation will cost $600. Cost of energy will average 3 cents per kilowatt-hour during the time the pump is used, but during the 7 months the pump is not operating a minimum charge of $50 per month is required. Both installations are to pay out over a 5-year period.

Comparative costs are as follows:

Source of power	Engine	Electric motor
Fuel cost, 0.43 lb fuel per bhp-hr, or 3,450 gal....	$ 207.00	
Lubricating oil, 2,000 rated hp-hr per gal........	15.00	
Maintenance, $1.50 per hp per year............	75.00	
Taxes, 1 per cent of equipment cost............	30.00	$ 6.00
Cost of electricity, 47,700 kwhr at 3 cents.......	....	1,231.00
Stand-by cost, 7 × $50......................	....	350.00
Fixed charges on equipment, 23.74 per cent.....	712.00	142.00
Total annual cost..........................	$1,039.00	$1,729.00

Fixed charges are determined from Table 12 on the basis that an equal payment will be made each year covering both the interest on the outstanding indebtedness as well as a portion of the principal. The table shows that a debt bearing 6 per cent interest can be retired in five annual payments, each payment being 23.74 per cent of the debt.

Example II. *Power for Radio Transmitter.*—A remotely located radio transmitting station requires a maximum electrical load of 60 kw and uses

Table 12.—Equal Annual Payments[a]

	2 %	2½ %	3 %		4 %	½ %	5 %		7 %	
	1.0200	1.0250	1.0300	1.0350	1.0400	1.045	1.0500	1.0600	1.0700	1.0800
	0.5150	0.5188	0.5226	0.5264	0.5302	0.5340	0.5378	0.5454	0.5531	0.5608
	0.3468	0.3501	0.3535	0.3569	0.3603	0.3638	0.367	0.3741	0.3811	0.3880
	0.2626	0.2658	0.2690	0.2723	0.2755	0.278	0.2820	0.2886	0.295	0.3019
	0.2122	0.215	0.2184	0.2215	0.2246	0.2278	0.2310	0.2374	0.2439	0.2505
	0.1785	0.1815	0.1846	0.1877	0.1908	0.1939	0.1970	0.2034	0.2098	0.2163
7	0.1545	0.1575	0.1605	0.163	0.1666	0.1697	0.1728	0.1791	0.185	0.1921
8	0.1365	0.1395	0.1425	0.1455	0.1485	0.1516	0.1547	0.1610	0.167	0.1740
9	0.122	0.125	0.1284	0.1314	0.1345	0.1376	0.1407	0.147	0.1535	0.1601
10	0.1113	0.1143	0.117	0.1202	0.1233	0.1264	0.1295	0.1359	0.1424	0.1490
11	0.102	0.1051	0.1081	0.1111	0.1141	0.117	0.1204	0.1268	0.1334	0.1401
12	0.0946	0.097	0.1005	0.1035	0.1066	0.1097	0.1128	0.1193	0.1259	0.1327
13	0.0881	0.0910	0.0940	0.0971	0.1001	0.1033	0.106	0.1130	0.1197	0.1265
14	0.0826	0.0855	0.0885	0.0916	0.094	0.0978	0.1010	0.1076	0.1143	0.1213
15	0.0778	0.0808	0.0838	0.0868	0.0899	0.0931	0.0963	0.1030	0.1098	0.1168
16	0.0737	0.0766	0.0796	0.0827	0.0858	0.0890	0.0923	0.0990	0.105	0.1130
17	0.0700	0.0729	0.0760	0.0790	0.082	0.0854	0.0887	0.0954	0.1024	0.1096
18	0.0667	0.0697	0.0727	0.0758	0.0790	0.082	0.0855	0.0924	0.0994	0.1067
19	0.0638	0.0668	0.0698	0.0729	0.0761	0.0794	0.082	0.0896	0.0968	0.1041
20	0.061	0.0641	0.0672	0.0704	0.0736	0.0769	0.080	0.0872	0.0944	0.1019
21	0.0588	0.0618	0.0649	0.0680	0.0713	0.0746	0.0780	0.0850	0.0923	0.0998
22	0.0566	0.0596	0.0627	0.0659	0.069	0.0725	0.0760	0.0830	0.0904	0.0980
23	0.0547	0.0577	0.0608	0.0640	0.0673	0.0707	0.0741	0.0813	0.0887	0.0964
24	0.0529	0.0559	0.0590	0.0623	0.0656	0.0690	0.0725	0.0797	0.0872	0.0950
25	0.051	0.0543	0.0574	0.0607	0.0640	0.0674	0.0710	0.0782	0.0858	0.0937
26	0.0497	0.0528	0.0559	0.0592	0.0626	0.0660	0.0696	0.0769	0.0846	0.0925
27	0.0483	0.0514	0.0546	0.0579	0.061	0.0647	0.0683	0.0757	0.0834	0.0914
28	0.0470	0.0501	0.0533	0.0566	0.0600	0.0635	0.0671	0.0746	0.0824	0.0905
29	0.0458	0.0489	0.0521	0.0554	0.0589	0.0624	0.0660	0.0736	0.0814	0.0896
30	0.0446	0.0478	0.0510	0.0544	0.0578	0.0614	0.0651	0.0726	0.0806	0.0888
31	0.0436	0.0467	0.0500	0.0534	0.0569	0.0604	0.0641	0.0718	0.0798	0.0881
32	0.0426	0.0458	0.0490	0.0524	0.0559	0.0596	0.0633	0.0710	0.0791	0.0875
33	0.0417	0.0449	0.048	0.0516	0.0551	0.0587	0.0625	0.0703	0.0784	0.0869
34	0.0408	0.0440	0.0473	0.0508	0.0543	0.0580	0.0618	0.0696	0.0778	0.0863
35	0.0400	0.0432	0.0465	0.0500	0.0536	0.0573	0.0611	0.0690	0.0772	0.0858
36	0.039	0.0425	0.0458	0.0493	0.0529	0.0566	0.0604	0.0684	0.0767	0.0853
37	0.038	0.0417	0.0451	0.0486	0.0522	0.0560	0.0598	0.0679	0.0762	0.0849
38	0.0378	0.0411	0.0445	0.0480	0.0516	0.0554	0.0593	0.0674	0.0758	0.0845
39	0.0372	0.0404	0.0438	0.0474	0.0511	0.0549	0.0588	0.0669	0.0754	0.0842
40	0.0366	0.0398	0.0433	0.0468	0.0505	0.0543	0.0583	0.0665	0.0750	0.0839
41	0.0360	0.0393	0.0427	0.0463	0.0500	0.0539	0.0578	0.0661	0.0747	0.0836
42	0.0354	0.0387	0.0422	0.0458	0.0495	0.0534	0.0574	0.0657	0.0743	0.0833
43	0.0349	0.0382	0.0417	0.0453	0.0491	0.0530	0.0570	0.0653	0.0740	0.0830
44	0.0344	0.0377	0.0412	0.0449	0.0487	0.0526	0.0566	0.0650	0.0738	0.0828
45	0.0339	0.0373	0.0408	0.0445	0.0483	0.0522	0.0563	0.0647	0.0735	0.0826
46	0.0335	0.0368	0.0404	0.0441	0.0479	0.0518	0.0559	0.0644	0.0733	0.0824
47	0.0330	0.0364	0.0400	0.0437	0.0475	0.0515	0.0556	0.0641	0.0730	0.0822
48	0.0326	0.0360	0.0396	0.0433	0.0472	0.0512	0.0553	0.0639	0.0728	0.0820
49	0.0322	0.0356	0.0392	0.0430	0.0469	0.0509	0.0550	0.0637	0.0726	0.0819
50	0.0318	0.0353	0.0389	0.0426	0.0466	0.0506	0.0548	0.0634	0.0725	0.0817
55	0.0301	0.0337	0.0373	0.0412	0.0452	0.0494	0.0537	0.0625	0.0717	0.0812
60	0.0288	0.0324	0.0361	0.0401	0.0442	0.0485	0.0528	0.0619	0.0712	0.0808
65	0.0276	0.0313	0.0351	0.0392	0.0434	0.0477	0.0522	0.0614	0.0709	0.0805
70	0.0267	0.0304	0.0343	0.0385	0.0427	0.0472	0.0517	0.0610	0.0706	0.0804
75	0.0259	0.0297	0.0337	0.0379	0.0422	0.0467	0.0513	0.0608	0.0704	0.0802
80	0.0252	0.0290	0.0331	0.0374	0.0418	0.0464	0.0510	0.0606	0.0703	0.0802
85	0.0246	0.0285	0.0326	0.0370	0.0415	0.0461	0.0508	0.0604	0.0702	0.0801
90	0.0240	0.0280	0.0323	0.0367	0.0412	0.0459	0.0506	0.0603	0.0702	0.0801
95	0.0236	0.0276	0.0319	0.0364	0.0410	0.0457	0.0505	0.0602	0.0701	0.0800
100	0.0232	0.0273	0.0316	0.0362	0.0408	0.0456	0.0504	0.0602	0.0701	0.0800

[a] Fish, John C. L.: "Engineering Economics," 2d ed., p. 249, McGraw-Hill Book Company, Inc., New York, 1923.

[b] n is number of years in period.

300,000 kwhr annually for operation of transmitters, lights, and station auxiliaries. The unit nearest the size required for maximum load is a 150-hp engine connected to a 100-kw generator. The cost of the entire plant containing two units is estimated to be \$23,000. Studies of load conditions indicate that 11 kwhr will be produced per gallon of fuel oil costing 5 cents per gallon delivered. Lubricating oil costs 50 cents per gallon, and maintenance is taken at \$1 per horsepower per year of installed capacity. The cost of the equipment is to be retired over a 10-year period with interest at 4½ per cent. The cost of energy delivered is developed as follows:

Energy delivered to radio station	300,000 kwhr
Energy generated (plant use 6.5 per cent of generation)	320,000 kwhr
Fuel cost $\frac{320{,}000 \times 0.05}{11}$	= \$1,460
Lubricating oil $\frac{150 \times 8760 \times 0.50}{2{,}500}$	= 263
Maintenance at \$1 per horsepower	300
Taxes at 1 per cent	230
Miscellaneous expenses	100
Fixed charges (Table 12, 12.64 per cent)	2,900
Total annual cost	\$4,253

No labor charge has been made against the operation of the plant since the operating staff of the station will attend to the operation and servicing of the equipment.

Example III. *Electric Generating Station.*—A small community needs an electric generating station for supplying its residents with light and power. The peak load during the year is 500 kw, and the total quantity of energy to be produced (including power for operating station auxiliaries) is 1,750,000 kwhr annually. The problem is to select the proper sizes and number of generating units to meet the requirements.

Supplying an electric utility calls for high-quality service. It must be continuous, and the plant must be capable of handling the load imposed upon it regardless of which of the generating units are available for service. It is necessary, therefore, to have installed in the station more generating capacity than that required to carry the peak load in order that any difficulty which might be experienced with one engine would not stop the plant from supplying the maximum demand. In this case it is necessary to carry a load of 500 kw regardless of which generating unit is out of service. There are many combinations of units that will produce this result, but we shall consider only two of them. These two combinations of units to provide firm capacity are as follows:

Plan A.—Two 500-kw generators connected to two 750-hp engines.

Plan B.—Two 330-kw generators connected to two 500-hp engines. One 170-kw generator connected to one 250-hp engine.

Under plan *A* either unit will handle the maximum load of 500 kw, while under plan *B* it will take two units to handle the maximum load.

Under some conditions, it has been proposed that a 500-kw generating unit and a 250-kw generating unit be installed for the conditions set forth in our illustration, owing to the fact that a large portion of the time the load is considerably below one-half the capacity of the 500-kw generating unit. Such a combination would be satisfactory if the requirements of utility service would countenance an occasional outage of considerable duration. Since an adequate plant must be capable of supplying the load which occurs, and since with the 500-kw unit down for adjustment or repairs, it would be possible to carry only a load of 250 kw with the smaller unit, such a proposed combination of generating units is not adequate to provide reliable electrical service. When such a combination is used for electric-utility service, the plant designer is merely gambling that it will not be necessary to have the 500-kw generating unit out of service at any time when a load in excess of 250 kw might occur. However, if a small unit is considered as a necessary addition to the plant in order to improve station over-all economy in the matter of fuel-oil consumption, then either three smaller units as proposed under plan *B* or two 500-kw units together with a unit of 170 to 250 kw are necessary.

The relative costs for the two plants as set forth under plan *A* and plan *B* are as follows:

Item	Plan *A*	Plan *B*
Installed capacity, kilowatts	1,000	830
Number of units	2	3
Total plant cost	$135,000	$143,000
Cost per kilowatts installed	$135	$171

The operation of either plant would require the same staff. As shown by Figs. 19 and 20, the maintenance, supply, and miscellaneous expenses would be practically the same for both installations, and for purposes of comparison let us consider that the maintenance costs for the three smaller engines would be about $300 a year less than for the station with the two larger units. Fixed charges would be based on 20-year bonds drawing 4 per cent interest.

By the method outlined in Chap. II, it is found that the use of the three smaller units as set forth in plan *B* will save 2,000 gal of fuel oil annually. The lubricating-oil consumption will be practically the same for either plant so that it can be neglected for purposes of comparing the two installations. With this data, and considering fuel oil costing 5 cents per gallon delivered at the plant site, it is now possible to make a comparison of the relative costs of these two proposed plans. These are summarized as follows:

	Plan *A*	Plan *B*
Total investment	$135,000	$143,000
Fixed charges (7.36 per cent)	9,928	10,525
Additional maintenance	300	
Additional fuel cost	100	
Comparative costs	$ 10,328	$ 10,525
Saving by plan *A*	$ 197	

Under plan *A*, where two units each of a size to handle the maximum load are used, the costs will be slightly cheaper than under plan *B* where three units are installed. There is a further advantage for plan *A* when considering an electric utility where the load is subject to future growth. As the load begins to increase, and it becomes necessary to add generating capacity, the installation of another 500-kw generating unit will raise the firm capacity of this station to 1,000 kw. In the case of plan *B*, when a fourth unit is added, the firm capacity of the station will be only 830 kw regardless of the size unit added equal to or exceeding 330 kw. It has been the experience of most small diesel-electric generating stations that when a small unit has been installed to carry the "valley load" from midnight till morning the unit was used only for a period of a few years since the constant growth in the plant load, including this valley load, necessitated the use throughout the 24 hr of the day of machines having greater capacity.

Industrial power plants present a somewhat different picture owing to the fact that the load growth will not be so great as that experienced in an electric utility. Furthermore, it is not necessary in many industrial establishments to have the continuity of

service necessary in a utility station. For example, many industrial establishments operate normally on either a 5- or $5\frac{1}{2}$-day schedule with the remainder of the week available for repairs and adjustments of equipment. Consequently, it is not always necessary to provide the stand-by capacity required in an electric utility where continuity of service is necessary. This point will be discussed further in the following chapter.

Each installation must be considered in the light of the limitations applying. No hard and fast rules are applicable to all cases, and the one making a study of the conditions for a particular installation must consider capacity of units, reliability necessary, as well as the economical unit or combination of units which will best meet the conditions.

CHAPTER V

THE INDUSTRIAL POWER PLANT

This chapter deals with the use of internal-combustion engines for prime movers in industrial and commercial establishments. Such plants are those serving a single establishment or organization in contrast to the utility plant that serves all consumers of electricity in a community or territory embracing many communities. In many respects the power plant for an industrial establishment is very similar to that required by an electric utility—in other respects it is vastly different. These similarities and differences form the basis of this chapter.

30. Where Engines Are Used.—Internal-combustion engines are employed as prime movers for electrical generating units which in turn produce the electricity required for the operation of industrial plants, stores, apartment buildings, and schools. They are also used for driving compressors, water pumps, cotton gins, and other mechanical-drive applications. In fact, the internal-combustion engine can be used as a power source for any service within the capacities of commercially produced engines.

They have been used in increasing numbers in industrial and commercial establishments over the past 15 years, owing to the fact that their use has been the economical solution to many power problems. Furthermore, the development of small- and medium-capacity units with rotative speeds ranging from 600 to 3,000 rpm has made tremendous progress. This development has provided engines of horsepower capacities at a price that has made their use attractive in many industrial, commercial, and institutional plants.

31. Design Problems.—The small plant for industrial or institutional service must be devoid of many of the auxiliaries and refinements necessary in central-station plants, owing partly to the smallness of the installation and partly to the limited funds available with which to provide an adequate power plant. With small units, many of the auxiliaries such as cooling-water

pumps, radiator for cooling jacket water, and fuel and lubricating-oil filters form a part of the engine assembly. Provisions, therefore, need not always be made by the plant designer for special cooling facilities or lubricating- and fuel-oil conditioning equipment as in the case of the large station.

Suitable housing facilities can usually be found in the building basement or in a room in the factory or institution without having to design a separate structure for the equipment. Even where separate housing facilities are required, the structural problems

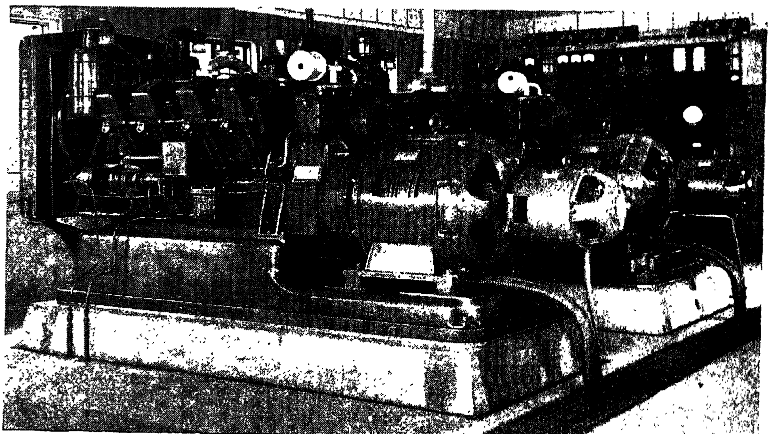

Fig. 27.—Diesel-electric units supply electricity at Old Faithful Inn, Yellowstone Park. (*Courtesy of Caterpillar Tractor Company.*)

involved are very minor in comparison to those encountered in the large station.

Electrical accessories necessary for small generating units are more or less standardized, and necessary switchboard panels and control wiring facilities are not complicated by many of the accessories that must be provided in large central-station installations.

The major problem of design in the small plant is the determination of those factors which are of prime importance; incorporating in the plant the necessary essentials and eliminating all unnecessary accessories and gadgets.

32. Primary Requirements.—Two items determine the design of the small plant. They are

1. Horsepower capacity required.
2. Type of service (continuous or intermittent).

Both these items will be considered somewhat in detail.

33. Horsepower Required.—The engine horsepower required in any instance must be determined from a study of available information concerning that particular case. If, for example, it is desired to drive a centrifugal pump with an engine, it is necessary to know the total pumping head, the quantity of water pumped, and the pump efficiency. From these data, the power requirement can be calculated by means of the equation

$$P = \frac{Q_w \times h}{3{,}960 \times e_p} \tag{1}$$

where P = horsepower input to pump.
Q_w = pumping rate, gallons per minute.
h = total pumping head, feet.
e_p = pump efficiency.

On the other hand, if an electrical load is to be supplied, the horsepower of the engine necessary to drive the generator can be taken for approximation purposes as 1.5 times the generator rating in kilowatts. Thus a 50-kw generator will require 50×1.5, or 75 hp of engine capacity to drive it. Standard sizes of a-c generators from 25 to 4,000 kw are given in Table 57, Chap. XVIII.

In each case a study of the individual conditions must be made to determine the power requirements. The study may require consideration of one or more of the following:

1. Amount of power being replaced.
2. Will the equipment being driven require the maximum, rated, or continuous output of the engine? (See reference to rating of engines operating over 750 rpm in Chap. VIII.)
3. Peak demand and characteristics of present electrical load (see Chap. II).
4. Analysis of manufacturer's ratings of power equipment in operation.

To determine this last item correctly, considerable care is required. If name-plate ratings of electrical motors, pumps, and

other power-consuming devices are the only sources of information in selecting a unit or units for providing power, the person making the estimate must be careful to see that the engine selected is not too small for the job nor yet so large that it never runs at full-load rating. In a study of this type it is necessary to know not only the aggregate horsepower required to drive all equipment, but also the horsepower normally operating and the maximum horsepower that will be operating for periods exceeding one-half hour, as well as the probable instantaneous maximum loads to which the engine will be subjected.

Consideration must also be given to the possibility of future load increases either by providing for extra capacity in the engine installed or by allowing sufficient space for the installation of more units at a later date.

34. Type of Service.—The relative continuity of service required from the small power plant will influence the number of units that must be installed, as well as the number and character of essential auxiliaries. The plant requiring power intermittently during a few months of the year would need only a single engine with a minimum of auxiliaries. The manufacturing establishment that is operating on a continuous 24-hr schedule 7 days a week would require a plant comparable to a central station in which at least two engines are installed with either engine being capable of carrying the entire plant load. Consider some actual examples dealing with the problem of service continuity.

Case I. *Cotton Gin.*—Cotton ginning is a seasonal business occurring during the cotton-picking season in late summer and early fall and usually extending over a 3- to 4-month period. Gins operate only when the planter brings his cotton pick to the mill. As a result, power requirements are both seasonal and intermittent. In a situation of this kind, it is not essential that power be available every hour throughout the year. A single engine of sufficient horsepower to drive the gin is all that is required. Plenty of time is available while the engine is not operating for required maintenance, and even though mechanical trouble might develop during the ginning season no serious damage to product in process would occur. A minimum of auxiliary equipment would be needed, and fuel storage would be held to a minimum consistent with requirements during the operating season and the ability to get fuel upon short notice. Obviously,

it would be unwise to end the operating season with a large fuel-oil stock in hand (say 10,000 gal) which during the following 8 or 9 months would be subject to loss through evaporation as well as being a potential fire hazard in a plant which is closed for a long period.

In a plant of this type where dust, cotton lint, and abrasive dirt permeate the air, special care must be taken to provide adequate cleaning of the air used by the engine.

Case II. Locker Plant.—The locker plant is a cold-storage plant equipped with individual compartments for the storage of

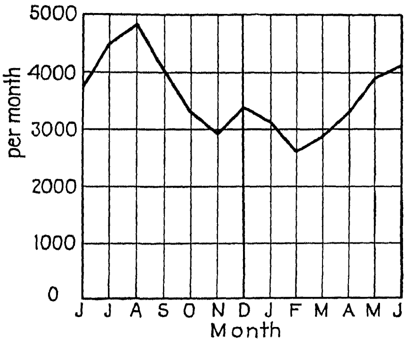

FIG. 28.—Electricity used by locker plant. (*Courtesy of Power.*)

meat and other food products. The plant staff prepares, freezes, and stores the food in individual lockers for future use by the customers. Mechanical-refrigeration equipment is employed, and the usual electrical demand for such a plant, including power for operating the refrigerating compressor, lights, and incidental power requirements, ranges from 15 to 25 kw. The electrical energy used by a typical locker plant is shown in Fig. 28.

An examination of the electrical load indicates that service is required throughout the year. Furthermore, because of the perishable nature of the foods stored, continuity of refrigeration is essential. The usual insulated-wall construction of locker plants, together with the stored refrigeration capacity in the system, permit extended shutdowns of the mechanical equipment (probably from 24 to 48 hr under favorable conditions) without damage to food stored therein. Some locker plants run their refrigeration

machinery only during the daytime and shut down for 8 to 12 hr at night.

Under such an operating schedule, normal engine and equipment maintenance could be done during shutdown hours. Furthermore, stand-by capacity in the form of a spare engine is not essential since any difficulty experienced with engines of a size required for a plant of this character could, in all probability, be remedied during a normal shutdown period.

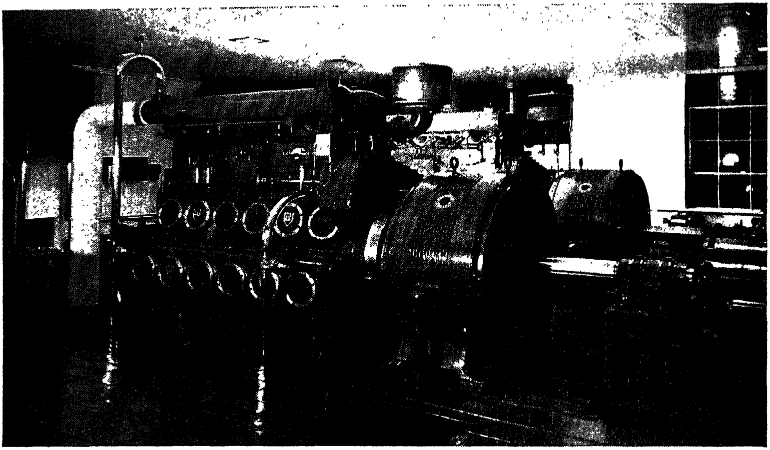

FIG. 29.—Diesels bring economy into the picture by handling peaks and summer loads at high efficiency at Connecticut College. (*Courtesy of Power.*)

In an installation of this type the engine could operate the refrigerating compressor with a belt drive, and, in addition, drive a small electric generator to provide the necessary lighting requirements. Under some conditions, it might be advantageous, although requiring a greater initial capital outlay, to provide two compressors and two engines, each providing somewhat over 50 per cent of the required refrigeration. With such an installation, the stoppage of either compressor or either engine for a prolonged period would not seriously interfere with the locker-plant operation.

Case III. *Apartment Building.*—The electrical energy requirements of an apartment building are in many respects very similar

to the load conditions experienced by an electric utility in a small town. Electrical energy must be supplied constantly for lighting, cooking, refrigeration, elevator, and other power services. Since power is required constantly, a sufficient number of suitable engine generating units must be available so that the failure of any unit would necessitate only a very brief shutdown or curtailment of service. A load condition of this type requires as a minimum the installation of two units, either one of which could carry the full electrical load in the building. Another operating plan would be to provide three units of such size that any two of them could carry the maximum load.

In a problem of this character, many factors must be considered, including consideration of available space for installation of generating units; investment involved in the installation of two, three, or perhaps four generating units; and effect of operating one or several engines in various combinations to improve fuel economy.

35. Combined Heating and Power.—Many industrial and institutional power plants require both heat and power either during certain seasons of the year or throughout the year. The tendency in such instances has been to provide a boiler plant and steam engine or steam turbine-driven generating units for producing the power and heat needed. Under such an arrangement, the steam used for power production is exhausted into the heating system. Where steam requirements exceed the engine or turbine exhaust, the excess is taken directly from the boiler through a pressure-reducing valve. If steam for power production exceeds that required for heating, the excess is often exhausted to atmosphere. Usually the demands for power and heat do not vary together; as a result there is considerable high-pressure steam delivered directly from the boiler to the heating system under some conditions, while exhaust steam from the engine or turbine must be wasted to atmosphere under other conditions. The plant operator desires to deliver both electricity and heat for the least total cost; consequently he is interested in any means that will permit him to keep them to a minimum.

There are many instances where one or more internal-combustion engines operating in conjunction with a steam power plant can supply excess energy requirements with the steam engines or turbines acting merely as reducing valves between the boiler and

the heating system. Power production by the steam units under this arrangement would be directly influenced by the rate of steam requirements for the heating load. Heat rejected by the internal-combustion engine to the cooling water and exhaust could be recovered in part for feed-water heating in the boiler plant. An example of such a combination as used in an institutional power plant is shown by the flow diagram, Fig. 30.

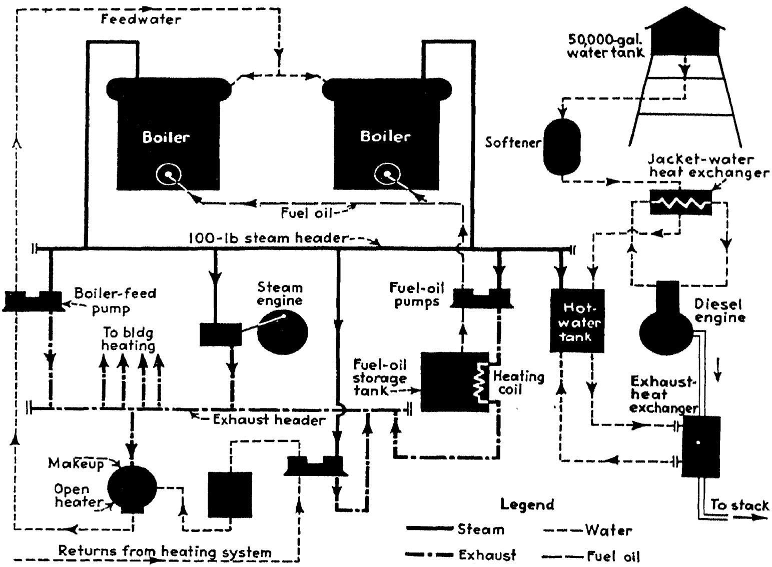

FIG. 30.—Flow diagram for combining diesel and steam equipment to provide an economical plant arrangement. (*Courtesy of Power.*)

Internal-combustion engines driving electric generators commercially fit into the industrial or institutional power plant supplying heat and power in many cases where

1. The heating load is seasonal, while power is required throughout the year.
2. The electrical load is in excess of that which could be produced through reduction in pressure of the steam for heating in an engine or turbine.
3. Heating steam is produced in the plant and electrical energy is purchased.

CHAPTER VI

PIPE-LINE PUMPING STATIONS

Transportation of petroleum products is becoming more important each year as the demands for gasoline, fuel oil, lubricants, and natural gas continue to increase. The bulk of refined petroleum products and natural gas must be transported a considerable distance from the wells to the point of consumption. It is necessary, therefore, that rapid, economical, and reliable transportation facilities be available. The pipe line has proved to be the best means of transportation where large quantities of petroleum products must be transported a considerable distance. Furthermore, the pipe line is the only practical means for conveying natural gas from the well to the distribution system in a community.

36. Historical.—The first pipe line built in the United States was a 2-in. line approximately 2½ miles long laid in 1862 from Tarr Farm to the Humboldt refinery at Plumer, Pa. The first trunk line was laid in 1875 consisting of approximately 40 miles of 6-in. line from the lower field of Butler County, Pennsylvania, to Pittsburgh, Pa. This was followed by a trunk line over 100 miles long from the same field to Cleveland, Ohio, in 1880.

The pipe line has demonstrated its economy and reliability for the transportation of natural gas, crude petroleum, and refined petroleum products; for over 320,000 miles of pipe lines are in operation in the United States, and the mileage has been increasing rapidly in recent years.

37. Types of Pipe Lines.—Pipe lines are usually classified as being in one of the following three groups:

1. *Crude-oil pipe lines* are those which gather the crude oil from the wells in an oil field and transport it to a near-by refinery or to a refining center located a considerable distance from the oil field.

2. *Gasoline pipe lines* are those which transport gasoline, kerosene, light fuel oils, butane, and propane from the refinery to

tidewater for boat loading or to centers of distribution which are usually hundreds of miles from the refinery.

3. *Natural-gas pipe lines* are those which transport natural gas from the wells to cities and towns where it is fed into the local gas-distribution systems.

The so-called gasoline pipe lines transport a wide variety of petroleum products. Heltzel[1] points out that, "a few years ago one would have questioned the practicability of transporting through a pipe line more than one class of product. Today, as many as eight different products representing several grades of gasoline, butane, distillate, tractor fuel and kerosene (covering the range of colors in the rainbow) are transported through a single long products line."

Such versatility has not been achieved overnight. Rather the advancement in the pipe-line art has been one of constant improvement in fabrication and installation of pipe, pumps, and engines for driving pumps.

38. Extent of Pipe Lines.—The mileage of pipe lines in the United States and Canada has increased from the first line 2½ miles long laid in 1862 to over 320,000 miles operated by 272 companies. The present mileage is divided between the three types of pipe lines as follows:

Pipe lines	Miles	Per cent
Crude oil	130,000	40.3
Gasoline	12,400	3.9
Natural gas	180,000	56.8
Total	322,400	100.0

Natural-gas pipe lines represent over one-half the total mileage, while gasoline pipe lines constitute the smallest portion of line miles.

The extent of individual systems varies widely for all three types of pipe lines. The greatest number of systems of any type are those transporting crude oil, while the most extensive pipe-line system transports natural gas. Data covering the extent and number of the various types of pipe lines are given in Table 13.

[1] HELTZEL, WILLIAM G., Future Trends in Pipe-line Practices, *Oil Gas Jour.*, Sept. 18, 1941, p. 82.

TABLE 13.—PIPE-LINE SYSTEMS

Type	Total line miles	Number of systems	Extent of system, miles		
			Maximum	Minimum	Average
Crude oil	130,000	133	12,380	1½	975
Gasoline	12,400	37	2,126	3	335
Natural gas	180,000	102	13,350	2	1,755
Total	322,400	272	13,350	1½	1,186

39. Oil-pipe-line Operation.—An oil-pipe-line system consists of the pipe connecting one or more pumping stations for forcing the liquid through it from the point of origin to its destination. The distance between stations varies from 30 to over 150 miles and is influenced by the capacity of the line, pipe diameter, topography of the country traversed, and characteristics of the fluid being pumped. Wolf[1] has treated this matter in considerable detail in his very excellent analysis of oil-pipe-line transportation systems. The methods of operating the pipe-line system have gradually evolved over the past 30 years until today they have reached a high state of perfection.

The original scheme of operation of a system having two or more pumping stations considered each section of line between two succeeding stations as a separate system. Each station following the initial one was provided with two operating tanks. The scheme of operation was to pump from the initial station *A* into one tank at the next downstream station *B*. When the first tank was filled, the incoming oil stream was diverted into the second tank. Pumping was then started from station *B* to the next downstream station *C*, suction being taken from the filled tank in station *B* and discharging into an empty tank at station *C*. This scheme of operation had the advantage of permitting close supervision of the quantity of oil in transit, and any line breakage or unusual losses were readily detected. The disadvantages of this system were the large investment required in tank storage at each pump station, possibility of contamination of the oil stream while in the pump-station tanks, relatively high evaporation losses, and inability to operate at full line capacity at all times.

[1] WOLF, OSCAR, Economic Design of Oil-pipe-line Transportation Systems, *Petroleum Mech. Eng.*, 1930, A.S.M.E., p. 30.

The difficulties encountered with the operation of the pipe-line system as a series of individual lines between stations led to the use of continuous pumping in which the variation in flow rate to and from each succeeding station was absorbed by means of a single storage tank connected on the suction side of each pump station. This is known as the over-and-short method of pumping. While this system was a forward step in that it required less tankage at each pump station, and permitted the line to work

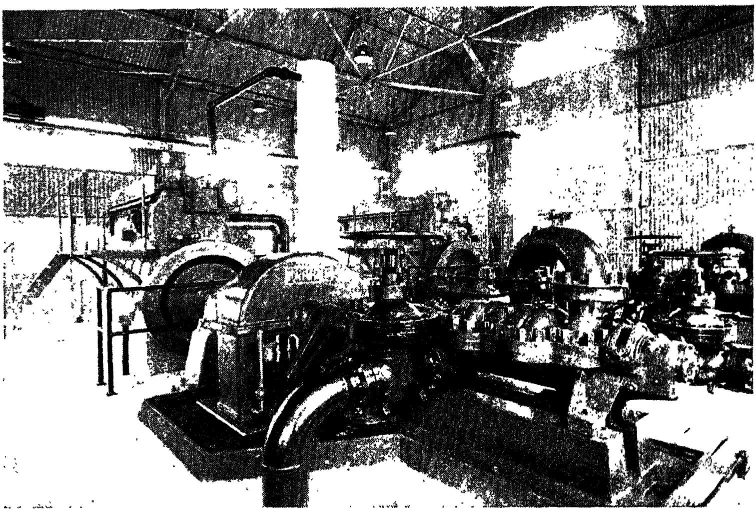

FIG. 31.—Oil-pipe-line pumping station, employing centrifugal oil pumps driven through geared speed increasers by diesel engines. (*Courtesy of The Cooper-Bessemer Corporation.*)

at higher over-all capacity, it was extremely difficult to keep an accurate account of the oil flow through the system. In order for the pipe-line dispatcher to keep track of the oil movement it was necessary that simultaneous readings of tank levels be made at all stations including those at both the initial and receiving points. This method of pumping also had the further disadvantages of possible contamination and relatively high evaporation losses.

In recent years, synchronized pumping has been used to considerable extent. With this pumping method, the entire pipe line

is considered as a unit, and the oil stream is pumped from a station as rapidly as it reaches it. Any variation in rate of flow to a given station causes a change in the rate of pumping to compensate for this variation. No tank storage is necessary at intermediate pumping stations except for such emergency storage as may be required; contamination is reduced to a minimum; the over-all capacity of the pipe line is greater than with either of the other methods of operation; and evaporation losses are reduced to a minimum. The operation of a system employing synchronized pumping involves rather careful manipulation of pumping equipment, particularly when starting up the line from a shutdown with the line full of liquid. Difficulties may be experienced in keeping pumps synchronized with line requirements, particularly where piston-type (positive-displacement) pumps are employed. Special metering facilities must be provided at each pump station to determine flow rate and quantity of oil pumped through the station.

40. Pumping Pressures.—Operating pressures for liquid pumps and gas compressors vary widely depending upon the construction and type of pipe line employed. For petroleum liquids transported any appreciable distance, pressures ranging from 600 to 900 psi have been common. Today the tendency is toward higher pump-discharge pressures[1] ranging from 1,000 to 2,000 psi.

Probably the first high-pressure crude-oil system placed in operation in the United States was the 8-in. line of the Utah Oil Refining Company from Fort Laramie, Wyo., to Salt Lake City, Utah, which went into service in 1939. Operating pressures range from 1,100 psi during the summer months to 1,450 psi, and occasionally 1,600 psi, during the winter. Crude oils are handled through this line. The pump station at Fort Laramie pumps the oil to Medicine Bow, a distance of 95 miles. In this distance the line rises 3,000 ft in its route across the mountains. Opal Station, pumping LaBarge crude oil through 27 miles of 4-in. tributary line, operated daily during the winter of 1940–1941 at pressures between 1,650 and 2,000 psi. The average pumping pressure was 1,850 psi.

41. Types of Pumping Units.—Both reciprocating (positive-displacement) and centrifugal-type pumps are used for handling

[1] Heltzel, *op. cit.*

petroleum liquids in pipe-line service. Reciprocating compressors are normally used for natural-gas pipe lines.

The *reciprocating pump* is built in several cylinder combinations, both single and double acting. In pipe-line service, usually duplex (two-cylinder) or triplex (three-cylinder) pumps are used. They are designed for large-capacity, high-pressure service. Because of the pulsating liquid flow, precautions must be taken to ensure that objectional surge pressures do not build up in the liquid line on the discharge side of the pump.

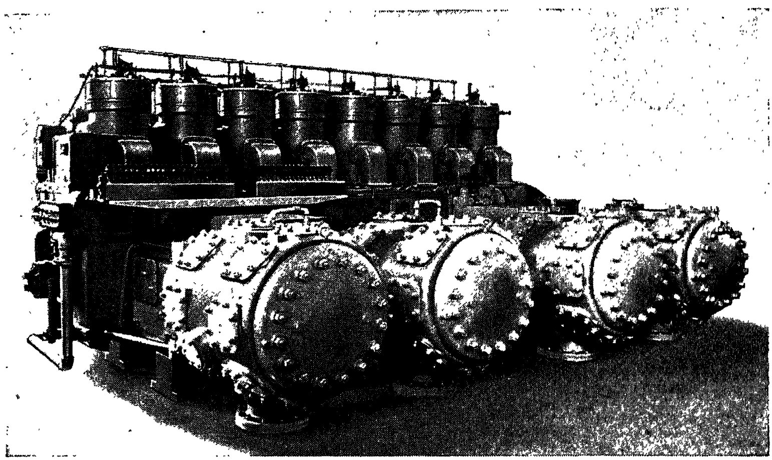

FIG. 32.—Angle compressor unit combining a vertical diesel engine and horizontal gas compressor. (*Courtesy of Clark Brothers Engine Company.*)

This type of pump has the advantage that at a given speed it will deliver practically a constant quantity of fluid regardless of the discharge pressure, being limited only by the stalling torque of the motor or engine drive. The pump discharge is not fully equal to the piston displacement, owing to some slippage of liquid past the piston. This slippage remains fairly constant for a given viscosity of oil over a fairly wide range of discharge pressures with the pump maintained in good operating condition.

The *centrifugal pump* imparts a high velocity to the liquid passing through it which in turn raises the pressure of the liquid at the pump discharge to a value set by the pump design. Unlike

the reciprocating pump, the variation in the discharge pressure of a centrifugal pump changes the rate of discharge of the liquid from the pump. This relationship between head and capacity of centrifugal pumps handling water is discussed in Chap. XII. Stepanoff[1] has shown that when a centrifugal pump handles petroleum liquids the head-capacity curve of the pump is made steeper, the pump efficiency decreases, and consequently the horsepower required to drive the pump increases as the viscosity of the liquid increases. The effect is shown in Fig. 33, when

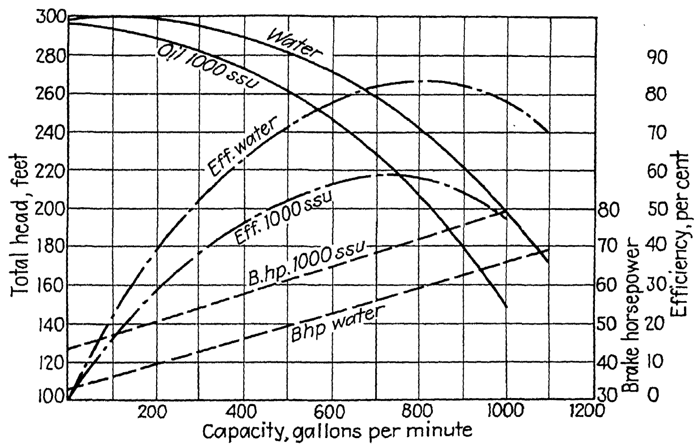

FIG. 33.—Effect of viscosity on head-capacity characteristics of centrifugal pumps.

comparing the same pump handling water or oil of a specific gravity 1.00 and a viscosity of 1,000 SSU.

42. Types of Prime Movers.—Internal-combustion engines burning either oil or gas and electric motors have been used for driving pumps in pipe-line stations. The internal-combustion engine is well fitted for this type of service since many pump stations are located at remote points where either electrcial service is not available, or the available capacity is insufficient to handle the pumping-station load. Fuel for engine operation is available wherever a pump station is needed on the line since it can be

[1] STEPANOFF, A. J., Pumping Viscous Oils with Centrifugal Pumps, *Oil Gas Jour.*, May 9, 1940, p. 78.

transported through the line as are other petroleum products or natural gas. Internal-combustion engines have a further advantage in that their speed is readily varied over a considerable range, a decided advantage in the operation of pipe-line pumps.

Electric motors are also widely used as prime movers for pipeline pumps. They are cheaper than internal-combustion engines

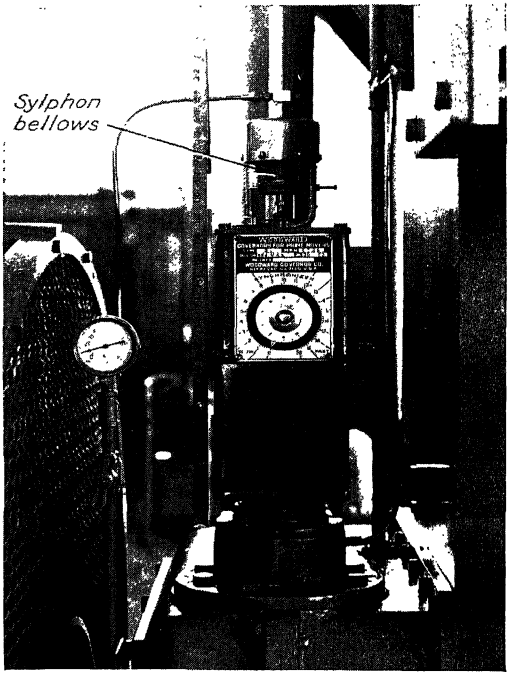

FIG. 34.—Woodward governor control for varying speed of diesel engine from suction pressure on positive-displacement oil-pipeline pump. (*Courtesy of Great Lakes Pipe Line Company.*)

of the same capacity and require considerably less space. The greatest disadvantage of the electric motors, particularly those operated by alternating current, lies in the fact that most of the motor designs are for constant-speed operation. Variable-speed a-c motors are available, although they are considerably more expensive than conventional units and the controls required for

their operation, particularly for those having a flexible speed variation, are rather complicated.

43. Engine Governing.—When internal-combustion engines are employed to drive reciprocating pumps on a pipe line utilizing synchronized pumping, it is necessary to provide one of the engines in each station with a special variable-speed governor control. In the operation of the pump station, all the engines, with the exception of the one provided with variable-speed governing, operate with their governors set to maintain constant rotative speed. The unit equipped for speed control automatically adjusts its speed to the variations in the rate at which the liquid arrives at the pump station. This is done by varying the speed so as to maintain constantly a predetermined pressure on the station suction header at all flow rates.

A hydraulic relay-type governor is used on the engine which is to operate at variable speed. A pressure regulator is mounted directly above the speed-changer spring of the governor. The sylphon bellows of this pressure regulator is connected to a second sylphon bellows on the station suction header by a small copper line filled with ethylene glycol (Prestone). Any variation in the pressure on the station suction header is immediately transmitted to the bellows in the pressure regulator on the engine-speed governor. If the pressure on the suction header drops, the variable-speed unit immediately slows down, while an increase of the suction-header pressure causes the engine to speed up.

44. Trends in Pipe-line Pumping.—The developments in the design of pipe lines over the past decade tend to indicate that lines will be operated at high pressures, probably as high as 2,000 psi, and with the trunk-line system operating from beginning to end as a close system in which all pump stations are synchronized. Variations in pumping rate will probably be obtained by varying the pressure on the line by the use of suitable pumps and variable-speed prime movers instead of employing spare-line capacity as is often done today.

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CHAPTER VII

THE POWER-PLANT BUILDING

Adequate housing facilities for the mechanical and electrical equipment are a necessity in any internal-combustion-engine power plant. If the equipment is to be installed in the basement of a hotel or apartment building, or within one of the existing structures forming part of an industrial establishment, the problem of securing adequate space for the equipment is no different from the problem confronting the plant designer who is providing new housing facilities for the utility power plant or the pipe-line pumping station. While this chapter is devoted largely to a discussion of the problems involved in the design of a new building for housing the engine equipment and auxiliaries, much of the material presented is applicable to any room or structure used for housing engine-driven equipment.

45. Basic Considerations in Building Layout.—The building used for housing engine-driven generating or pumping equipment should be designed with its function and use kept constantly in mind. While this appears self-evident, it is surprising to find many power-plant buildings which apparently have been designed to create a beautiful exterior and with no attention paid to the arrangement of equipment in the interior, or to the possibility of having to increase the size of the building in the future to house additional and probably larger units. Any power-plant building which is adequately arranged for its main purpose must be so designed that additions can be made to it with a minimum of changes in the portion already constructed.

In order to accomplish the purpose for which the power-plant building is intended, so-called *modern architecture* in which a pleasing appearance is obtained through the disposition of building mass and line rather than through the use of an excessive amount of ornamentation is generally found to be most advantageous. Thus simplicity in design together with the provision of adequate space in the engine room become the basic considerations in the layout of the plant structure.

The building design should start with a layout of the operating floor. The equipment is located first, and the building is then planned around it rather than the reverse where the building is designed and the equipment fitted into it. Considerable care should be taken in arranging equipment in such a manner that additions can be made with a minimum of change. Thus it becomes necessary early in the design of the plant to decide upon the spacing required between center lines of adjacent units, the distance from the center line of the end unit to the end wall, and the distance required between the head end of the engine and the wall, and the generator and its adjoining wall.

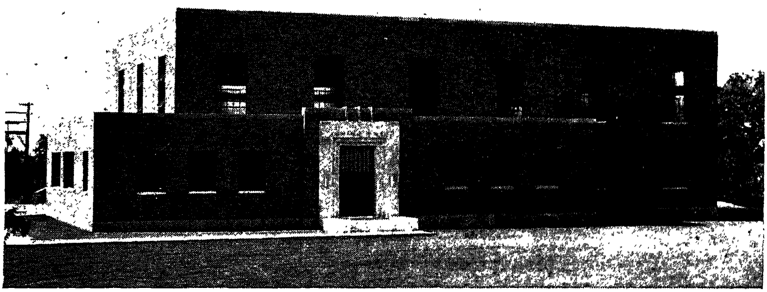


FIG. 35.—Modern diesel power-plant building, designed to house five units. (*Courtesy of Burns & McDonnell Engineering Company.*)

An example of careful planning of a plant is shown in Fig. 36. This is the floor plan of the building shown in Fig. 35 in which four units are now installed and provisions have been made for space to accommodate a fifth engine at a later date. In this layout the building was arranged for additions to the right. All units were placed parallel in the engine room in order to provide the shortest possible electrical circuits from the generators to the switchgear as well as short air-intake and exhaust lines. Provisions were made for bringing equipment into the plant at the left, and the dock ramp was designed to permit backing trucks into the building and removal of equipment from the truck with the plant crane. The design of this particular plant is somewhat unusual in that all floors in the engine room, which

is 52 by 96 ft, are cantilevers. Such construction permits removal and replacement of units in the engine room with a minimum of disturbance to existing construction. Sectional views through this plant are shown in Figs. 37 and 45.

Unfortunately all plants do not receive the careful planning shown in the plant just mentioned, as for example the plant for which Fig. 38 is a floor plan. While space has been provided for

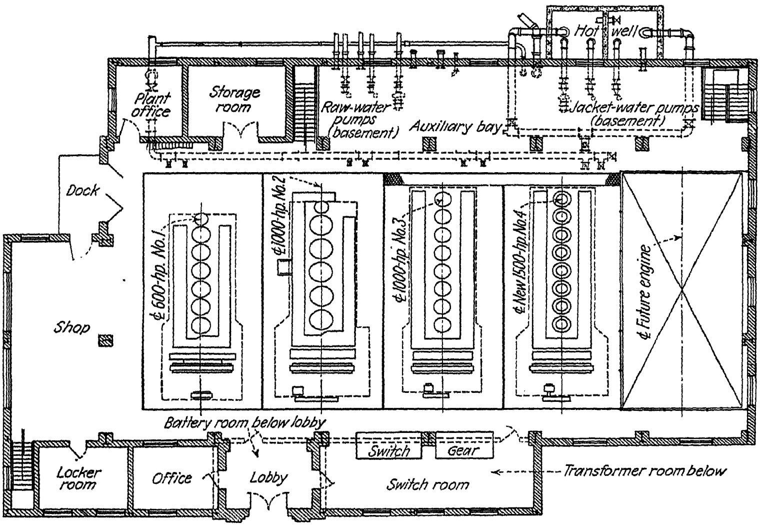


Fig. 36.—Floor plan of power plant shown in Fig. 35.

a fourth unit in the layout, the size of the machine that can be installed in this space is limited.

If the next unit needed is larger than can be accommodated in the space provided, then changes requiring major building alterations are necessary. In this particular case, the probable addition for a larger unit would have to be made either to the right or left of the present engine room. When the addition is made in either direction, existing air intakes and air-intake piping must be changed, requiring major alterations in the present building. The generators and exciters are so far removed from the switchboard that the cost of wiring is more than necessary, and the length of exhaust lines from the units nearest the office

and storeroom are longer than would have been necessary if the units had been arranged as shown in Fig. 36.

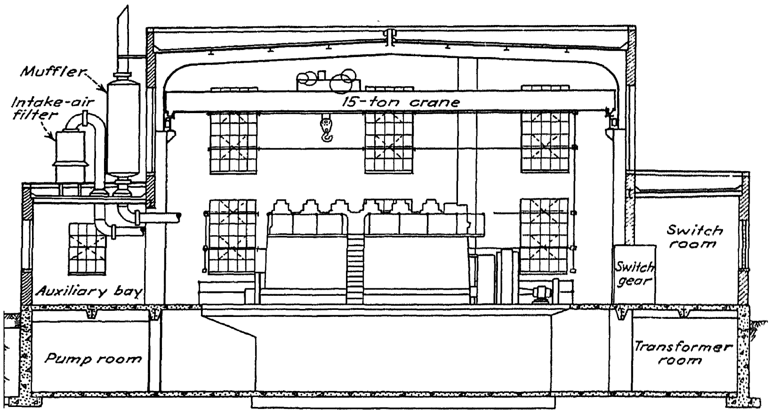


FIG. 37.—Cross section through plant shown in Fig. 35.

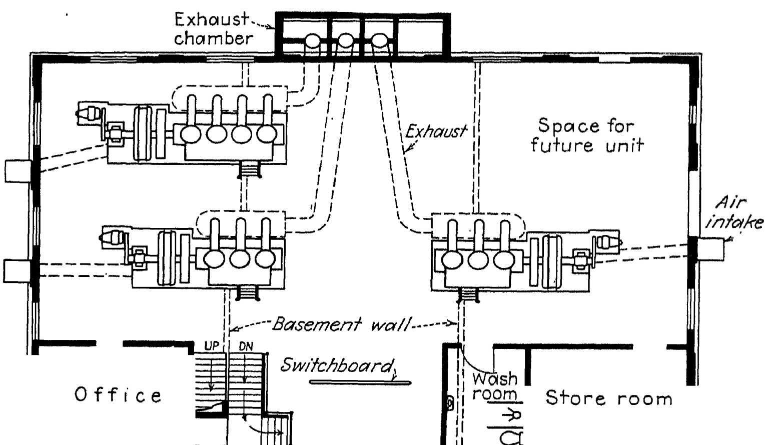


FIG. 38.—Floor plan of plant not readily expanded to house more than four units.

In view of the possibility that much larger units may be required at a later date in any electric generating station, the advisability of providing space in the original building for future

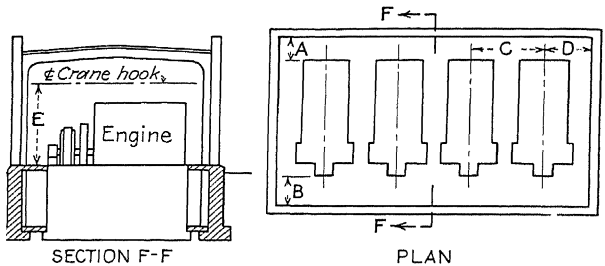
generating units is often open to question. There are conditions under which it may be justifiable to provide space for future units at the time of construction of the original power-plant building, but these do not occur in the majority of plants designed. In general, additional building space should be provided only in those cases where the capacity of the future units is definitely known. This is seldom known with any degree of certainty.

46. Space Requirements.—The space required for engine equipment is influenced by several factors including capacity of units, rotative speed, design of electric generators employed, foundation properties of the soil upon which the plant is constructed, and the desires of the operating staff for special features of plant arrangement which may greatly increase the size of the plant building. The best criteria for spacing engines in any plant are derived from one's own observations of both good and bad plant layouts. As an aid in this direction, data from actual plants are summarized in Fig. 39, showing spacing of units of various sizes and rotative speed. While these data are taken from plants produced by engineers well versed in diesel-plant design, they are not to be considered as applying in every instance. Many limiting factors had of necessity to be considered in each instance, and it will be found that in every plant layout limiting conditions peculiar to the individual case influence the spacing of generating units as well as space requirements for auxiliaries.

The effect of rotative speed on the spacing of engine units can be shown by comparing the recommended spacing between center lines of units of approximately the same horsepower designed for operating at different revolutions per minute. Thus, while 10 ft between engine center lines for units operating at 400 rpm is recommended by one engine manufacturer, a distance of 15 ft is recommended for units designed to operate at 225 rpm. This variation in spacing is influenced somewhat by the design of the generator installed, since the physical dimensions of generators for the same capacity increase with a decrease in rotative speed.

Spacing of units may be affected by the bearing capacity of the soil upon which the foundations rest, since in dealing with some soils it is necessary to increase the area of the foundation in contact with the soil in order to reduce the bearing pressure per unit area to a safe value.

The desire of the plant superintendent or operating staff for special operating features may have a decided bearing upon the spacing of equipment in the plant. Thus the incorporation of special shop facilities, instrument panels incased in building walls with attendant structural additions to house instruments, or unusual combinations of fuel- and lubricating-oil facilities installed in the power-plant basement may involve the inclusion



Engine rating			Dimensions					Remarks
hp	rpm	App. kw	*A*	*B*	*C*	*D*	*E*	
450	300	300	12′-6″	5′-0″	15′-6″	10′-6″	15′-0″	No traveling crane; hoist on roof truss
600	400	400	10′-0″	6′-0″	13′-0″	9′-0″	18′-6″	No basement; hoist on roof truss
800	200	575	9′-6″	7′-0″	16′-0″	13′-0″	21′-0″	Old design of engines requiring greater clearances
1,000	225	700	13′-0″	9′-0″	16′-0″	12′-0″	16′-6″	
2,250	225	1,500	13′-6″	6′ 6″	24′-0″	16′-3″	27′-0″	Plant designed to accommodate 4,000-hp units

Fig. 39.—General dimensions of internal-combustion-engine power plants.

of more building volume than normally required. In planning the layout of station switchgear, adequate space around the gear should be provided. Where the open-type switchboard is installed, at least 8 ft should be allowed between the front of the switchboard and the wall behind it in order that maintenance of breakers and accessories may be done in safety. Where metal-clad switchgear is used, sufficient space must be provided behind the gear for making connections, repairs, or removing equipment as required for the gear installed.

47. Building Details.—In most cases the construction of a basement in a diesel power-plant building is warranted. Foundation blocks for engines are constructed at least 6 ft deep. Since the engines take up a considerable proportion of the floor space on the ground floor of the building, a large portion of the excavation necessary for an adequate basement is required for foundations regardless of whether or not the basement is constructed. While the construction of a basement in a diesel plant does involve additional cost in reinforced-concrete basement walls and operating-floor sections, the additional space provided for the location of plant auxiliaries is generally worth the additional expense. If the basement is not constructed, care must be taken in backfilling around the foundation blocks for the engines in order that the operating floor will have solid support. In general, plants constructed without basements require a relatively greater floor area for the same generating capacity than do plants constructed with basements.

The design of a basement in the power plant calls for careful planning in order that equipment will be arranged properly and sufficient space be made available clear of columns necessary for supporting the operating floor. The depth of the basement should be such that a man can walk upright underneath exhaust and air-intake piping which may be run close to the basement ceiling. In order to do this, it is advisable to provide at least 10 ft from the basement floor to the operating floor. In some cases it may be necessary to increase this height in order to obtain the necessary working headroom, particularly since it is desirable to carry all plant piping just below the operating-floor slab in order to keep passageways between items of auxiliary equipment located in the basement free of interfering piping.

In many instances, the operating-floor slab can be built largely as cantilever sections supported either from the engine foundations or the reinforced-concrete walls of the basement. Such construction has the advantage of eliminating much framing in the basement consisting of beams, columns, and other supporting members which interfere with the location of piping and auxiliaries in all too many instances. The secret of such design lies in the use of reasonable cantilever lengths and adequate design of the reinforced concrete together with careful placing and mixing of the concrete during its installation.

In all too many cases, sufficient thought has not been given to elevating the crane a sufficient height in the power plant so that larger engines can be installed in the future and be serviced with the crane provided in the original installation. The reason that cranes are often set too low is that the architectural treatment of the building dictates a definite wall height for good exterior appearance. With a set wall height to give a pleasing appearance, the building designer allows 2 to 3 ft for a parapet wall and

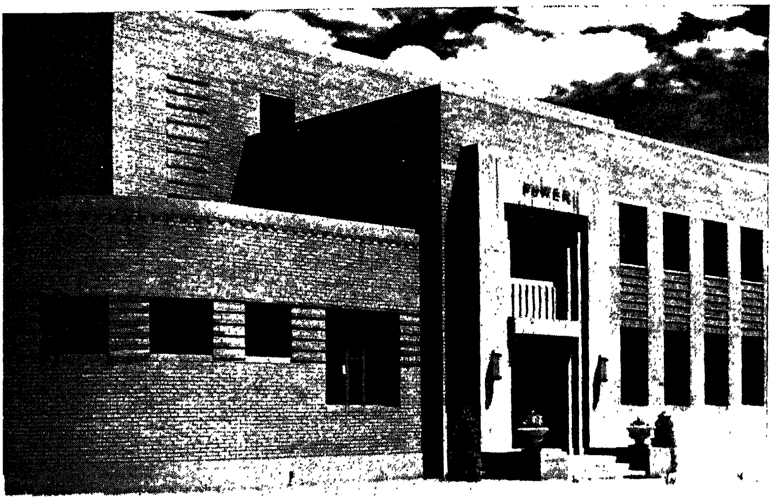


FIG. 40.—Modern treatment of a diesel power-plant building can be very pleasing. (*Courtesy of Burns & McDonnell Engineering Company.*)

4 to 6 ft for the depth of a roof truss. Since most crane manufacturers require 12 to 18 in. clearance between the top of the crane and the underside of the roof truss and the crane sets 2 to 3 ft above the crane rails, these various allowances for most plant buildings range from 9 to 14 ft. Thus, the crane rail in most conventional building designs is a considerable distance below the top of the exterior wall line.

If this distance of 9 to 14 ft is to be reduced, the conventional roof-truss construction must be eliminated and a method of framing utilized for the roof which does not take up so much

space. The *rigid-frame construction,* Fig. 37, in which the supporting columns set in the building walls and the roof beams are welded together to form a continuous section, permits the elimination of the conventional truss construction for supporting the roof and saves 2 to 4 ft of vertical distance usually taken up by this truss. Such construction is much to be preferred because it improves the appearance of the interior of the plant and permits the raising of the crane without increasing the wall height.

48. Materials of Construction.—Materials employed for the creation of a power-plant building vary widely, ranging from sun-dried adobe brick to reinforced concrete, depending upon the locality and severity of weather conditions encountered. Types of construction include timber framing with corrugated wall and roof covering; structural-steel framing covered with metal or transite sheets; reinforced-concrete substructure with structural-steel and brick superstructure; reinforced-concrete substructure with native-stone or brick superstructure and without steel framing; and structures composed entirely of reinforced concrete.

Each type of construction has been used where its particular characteristics fit the local requirements. For example, in the desert country of the southwestern section of the United States the use of heavy timber or structural-steel framing covered with corrugated metal or corrugated transite is rather common. The needs for heating are not great, metal corrosion difficulties are a minimum, and the large building volumes needed during the extremely hot summers for cooling and ventilation can be provided much more economically with this type of structure.

For most plants, however, and particularly for those in the northern half of the United States, the use of reinforced-concrete substructures and brick and structural-steel superstructures are the rule, with buildings constructed entirely of reinforced concrete coming into the picture. Between the two types of buildings, there is probably little difference in cost. The brick and steel superstructure has the advantage of permitting alterations much more readily than does the reinforced-concrete superstructure. Extreme care must be exercised in the construction of the reinforced-concrete building to ensure watertightness of walls and elimination of vibration transmission from the engines. If the wall structure is not watertight, action of the elements, particularly freezing and thawing in the northern regions, will tend to

disintegrate reinforced-concrete walls and result in extensive repairs being required periodically.

49. Roof Construction.—Several types of materials are available and have been used for the construction of power-plant roofs. The cheapest roof material is wood with composition roofing material applied over it. While the cheapest in first cost, it is

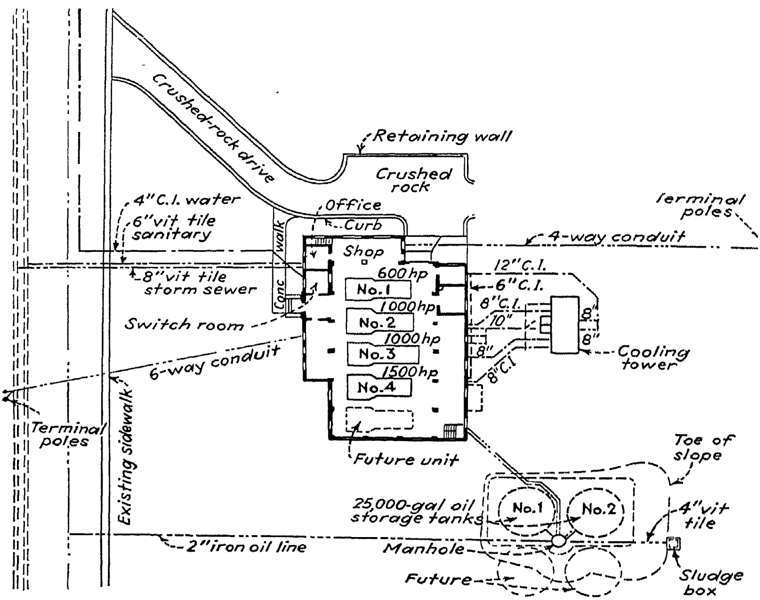


FIG. 41.—Layout of grounds for a diesel-electric plant requires considerable care. (*Courtesy of Burns & McDonnell Engineering Company.*)

open to the objection that it is not fireproof and very probably will be more expensive to maintain than other types of roof construction. The most expensive type of roof is that constructed of reinforced-concrete slabs which are in turn covered with ½ to 2 in. of insulating material and finished with a tar and gravel topping. It is the most permanent type of roof, is fireproof, and the maintenance cost is very low. Formed steel sections are also used for roof construction. These sections are bolted or welded to the roof purlins and then covered with insulating material and a

tar and gravel topping surface. While somewhat cheaper than the reinforced-concrete slab construction, it is considered one of the better types of roof construction available for power-plant purposes.

Corrugated transite and corrugated metal are also used for roof construction, but their use has largely been restricted to power plants and pipe-line pumping stations operating in the southwestern part of the United States.

50. Windows and Doors.—In general, metal window sash are preferable for power-plant construction on account of their fireproof and service qualities. They can be obtained in a wide variety of styles in either steel or aluminum, although the steel sash is used much more extensively than its aluminum counterpart in power-plant construction. Sash are designed to use either 12- by 18-in. or 14- by 20-in. standard glass panels and are arranged for any grouping of units desired. The smaller glass size is generally preferred. Hinged sections can be provided for ventilating purposes wherever required in the window. They can be operated either with pull chains or with special mechanical operators of which there are a number on the market.

Where window screens are not required over ventilator sash openings, horizontally pivoted sash are usually installed because they are the cheapest type available. Ventilated sash requiring screening should be of the projected type which is more expensive and better constructed than the horizontally pivoted type. Screens should be installed on the outside with the projected sash hinged at the bottom and opening into the building to secure the simplest workable arrangement.

Metal doors are used extensively because of their excellent fireproof and service qualities. Hollow metal doors are usually employed for main-entrance, office, and other doors where appearance as well as serviceability count, while tubular-steel or industrial-type doors are most frequently used for those openings such as storeroom, service entrance, shop, and doors in places where appearance is of secondary importance. Class *A* fireproof doors are generally used for openings into transformer vaults, although such doors are not absolutely required in a building used solely for power-plant purposes.

In those places where large openings must be provided through exterior walls for moving in large pieces of equipment,

of doors are available. These are the rolling steel door and the swinging multileaf tubular-steel door. The rolling door is the more easily handled in opening and closing, but it is not favored in many cases because it does not always fit tight enough to keep wind, rain, and snow from entering the building around the outside vertical edges of the door, and occasionally through

TABLE 14.—SASH DIMENSIONS STANDARD WITH MOST MANUFACTURERS

Number of sash wide	Over-all sash width using	
	12- by 18-in. panes	14- by 20-in. panes
2	2 ft 1⅝ in.	2 ft 5⅝ in.
3	3 ft 2 in.	3 ft 8 in.
4	4 ft 2⅜ in.	4 ft 10⅜ in.
5	5 ft 2¾ in.	6 ft 0¾ in.
6	6 ft 3⅛ in.	7 ft 3⅛ in.

Number of sash wide	Over-all sash height using	
	12- by 18-in. panes	14- by 20-in. panes
1	1 ft 7¼ in.	1 ft 9¼ in.
2	3 ft 1⅝ in.	3 ft 5⅝ in.
3	4 ft 8 in.	5 ft 2 in.
4	6 ft 2⅜ in.	6 ft 10⅜ in.
5	7 ft 8¾ in.	8 ft 6¾ in.
6	9 ft 3⅛ in.	10 ft 3⅛ in.
7	10 ft 9½ in.	11 ft 11½ in.

NOTE.—The opening for a sash two panes wide and six panes high using 12- by 18-in. panes is 2 ft 1⅝ in. by 9 ft 3⅛ in. Larger windows than provided in this table can be secured by assembling several standard windows by means of muntin bars.

the interlocking leaves of the door. The large multileaf tubular-steel door can be sealed against wind, rain, and snow better than can a rolling type of door. Consequently it is often preferred by plant operators even though it may not be opened and closed so readily as the rolling type.

In using swinging doors for large openings, it is found convenient in many cases to install removable transoms, thereby decreasing the height of the doors necessary. Since the full size of the opening is required only at infrequent intervals for

moving equipment, the objections to removing a transom are more than offset by the greater ease with which the smaller doors can be handled. Where the size of the swinging doors is unwieldy, a small access door can be installed in one of the large doors for use in normal plant operation. Access doors can also be installed in rolling doors, although the construction is somewhat complicated.

TABLE 15.—STOCK SIZES SWINGING INDUSTRIAL-TYPE DOORS

Single doors		Double doors	
Width	Height	Width	Height
2 ft 6 in.	7 ft 0 in.	5 ft 0 in.	7 ft 0 in.
3 ft 0 in.	7 ft 0 in.	6 ft 0 in.	7 ft 0 in.
3 ft 6 in.	7 ft 6 in.	7 ft 0 in.	7 ft 6 in.
4 ft 0 in.	8 ft 0 in.	8 ft 0 in.	8 ft 0 in.
5 ft 0 in.	10 ft 0 in.	10 ft 0 in.	10 ft 0 in.

NOTE.—Hollow metal doors are built to order, and consequently no standard sizes are given for this type of door.

51. Floors.—The construction of floors in a power plant presents a problem that must be given more than ordinary consideration. Several types of materials are available with which to construct a satisfactory floor, and the type of material used will largely determine the method of its installation as well as the time of installation during the construction program. In general, power-plant floors are built of reinforced concrete, and this material is preferred in most instances. It is the general practice, where a basement is constructed in the building, to place the rough slab for the main operating floor prior to the installation of equipment on the foundations. After the equipment is installed, the topping course of concrete is then applied. When this construction procedure is followed, extreme care must be taken to see that oil and other dirt is not permitted to stain the surface of the rough floor slab, or difficulties will be experienced in obtaining a satisfactory bond between the rough floor and the finish when it is poured. Some engineers prefer to construct the floor complete, including the finish course, before any equipment is installed in order to ensure satisfactory bond between the courses, although this construction method has the disadvantage

of requiring very careful protection of the floor surface during the erection of equipment to eliminate undue damage to it.

Often the final concrete course is tinted by means of pigment to the color desired. Several types of coloring materials are available which when mixed with the final surface course give a very satisfactory floor color. Special floor dyes are also available for coloring the concrete after the floor has been placed.

In many instances it is desired to finish the floor with quarry tile. When such topping surface is used, it is advisable not to place the tile until after the engines and their accessories have been erected. When such floor finish is employed, care must be taken to see that the rough floor surface is not stained by oil or other materials that will prohibit satisfactory bond between the setting grout used with the tile and the rough floor slab. If bond is not secured, loose tiles will create an unsatisfactory floor surface.

Where a basement is not provided, it is possible to do all the backfilling around the engine foundations and erect the engines and accessories prior to the construction of the floor slab. When equipment erection has been completed, the reinforced concrete slab can then be poured and the finish course applied without the danger or possibility of poor bond between the rough and finish courses.

In rare instances, the floor may be constructed of creosoted block. Such material makes a satisfactory floor, but it is open to the objection that unless an adequate concrete subfloor is constructed under it, the blocks settle unevenly and produce a very unsatisfactory floor surface. Most plant designers and operators feel that if a concrete subfloor is necessary, the entire floor should be constructed of concrete.

52. Crane Facilities.—A crane for handling cylinder heads, pistons, connecting rods, and other parts of internal-combustion engines is a necessity in any well-planned power plant. Except in the case of small high-speed engines, all parts normally handled during maintenance operations require the use of hoisting facilities. Too often these facilities available for maintenance purposes are entirely inadequate both from the viewpoint of lifting capacity and mobility.

The most usable type of hoist is the traveling crane which permits lifting anywhere within the limits of the crane runway, which

in most instances is the length of the engine room. Where funds are not available to provide a suitable traveling crane and crane rails, a fairly satisfactory substitute is a single-rail hoist located on the center line of each engine unit which permits removing engine or generator parts and lowering them to the floor either in the aisle space at the front of the engine or at the rear near the exciter. Since the most usable space for placing engine parts during an engine overhaul is the aisle space between adjacent engines, this single-rail hoisting arrangement requires that parts be lowered onto a hand truck, moved to the aisle space between units, and removed from the truck by manpower.

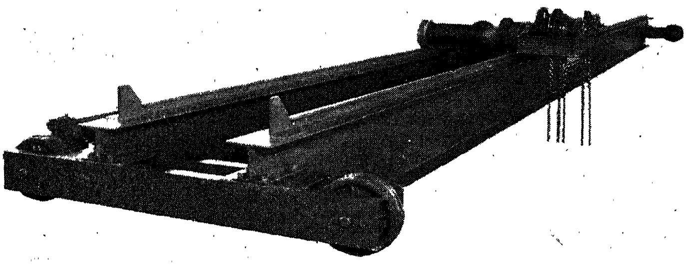


Fig. 42.—Hand-operated, double-girder crane of 5-ton capacity. (*Courtesy of Shaw Box Crane & Hoist.*)

The investment necessary for a suitable traveling crane and crane rails is generally found advisable where several engines are to be maintained. The investment in the crane is determined largely by the necessary span and the type of operating mechanism employed for the hoist, bridge, and cross bridge, and is not greatly influenced by the lifting capacity for the sizes generally used in internal-combustion-engine plants. In general, cranes of 15 tons capacity and less are provided with manual operation throughout, while cranes above 15 tons capacity are generally provided with a motor-operated hoist and manual bridge and cross-bridge travel. Many operators prefer that the hoist be motor operated in order that the possibility of scoring pistons or other fitted parts due to the rubbing of the hoisting chain against them will be eliminated. It is seldom that complete

motor operation of the crane is used except in large stations, or where an all-electric crane is found necessary.

Where a hand-operated crane is used, the hand chain for operating the lifting hook should be offset to eliminate rubbing of the chain on parts being raised or lowered. In general, a crane with a double-girder bridge is preferable to that with a single girder, particularly for long spans and heavy loads.

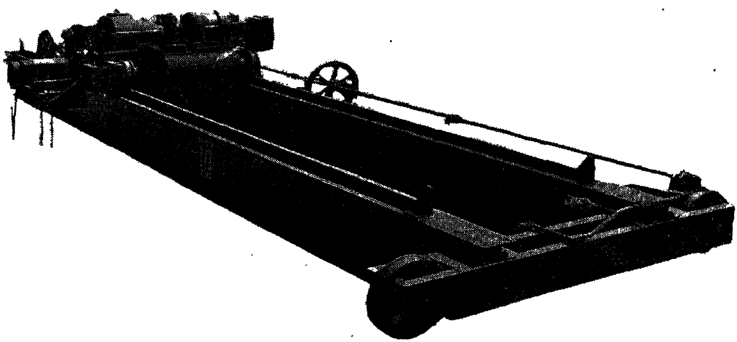


FIG. 43.—Motor-operated, double-girder crane of 10 tons capacity. (*Courtesy of Shaw Box Crane & Hoist.*)

53. Foundations.—Internal-combustion-engine foundations are often built in accordance with plans furnished by the engine manufacturer. The engineer designing the plant should use such standard foundation plans as a guide only and should employ his knowledge of soil mechanics, reinforced concrete and structural design, and mechanics of the engine under consideration to evolve a foundation satisfactory for the job under consideration.[1]

The first information required in starting the design of a diesel-engine foundation is the character of the soil upon which the foundation is to be placed. Borings should be made on the site of the foundation to determine the type of soil upon which the foundation is to rest. Solid rock, caliche or cemented gravel, firm clay, and confined gravel in general provide excellent support for engine foundations. Caliche is open to the objection that it

[1] LARKIN, KENNETH H., Design of Diesel-engine Foundations, *Trans. A.S.M.E.* vol. 64, No. 4, p. 341, May, 1942.

transmits vibrations. This objection can be mitigated by providing a sand cushion between the foundation block and the excavation in the caliche. Some clays, sands, and water-bearing gravel together with bog or marsh lands are not good foundation materials. When it becomes necessary to locate equipment on such subsoils, special precautions must be taken to ensure an adequate foundation. Piling footings, spread footings, and other precautions must be taken under such conditions. More than anywhere else in the design of a diesel plant, the analysis of the condition of the soil upon which an engine and its foundation are to rest and the means of obtaining proper foundation bearing should be the work of an expert in soil mechanics.

After the characteristics of the soil upon which the foundation is to be placed are determined and a satisfactory loading per square foot of ground area for the particular soil has been set, the next problem is to determine the weight of foundation necessary and the manner of distribution of this foundation mass under the engine and the equipment driven by it. The final decision on this important point should be reached after conferring with the manufacturer of the particular engine for which the foundation is being prepared. Considerable difference of opinion exists among engine builders as to the weight of foundation which should be provided per rated horsepower of engine capacity. Since the number of cylinders, rotative speed, stroke cycle, unbalanced moments, and unbalanced forces all have a bearing upon the problem, the development of foundation-weight data applicable to all engines is impossible. Table 16 gives the approximate foundation volumes for engines running at 400 rpm and less.

TABLE 16.—AVERAGE VALUES OF FOUNDATION YARDAGE

Number of cylinders						
Foundation volume, cubic yard per brake horsepower	0.141	0.120	0.108	0.100	0.096	0.091
Number engines tabulated	7	10	20	20	14	14

Engine foundations should generally be isolated from the building proper in order to eliminate or reduce vibration transmission. In order to do this, the entire substructural design of the plant must be considered as a unit. Basement floor, engine founda-

tions, and the operating-floor design should be coordinated to prevent vibration transmission as well as provide floors of sufficient stiffness and strength to withstand the loadings which may be placed upon them. By the use of suitable plastic filling materials, it is possible to obtain a satisfactory isolation between the engine foundations and the basement floor. Complete separation of the foundations from the operating floor must also be accomplished in order to eliminate the possibility of vibration

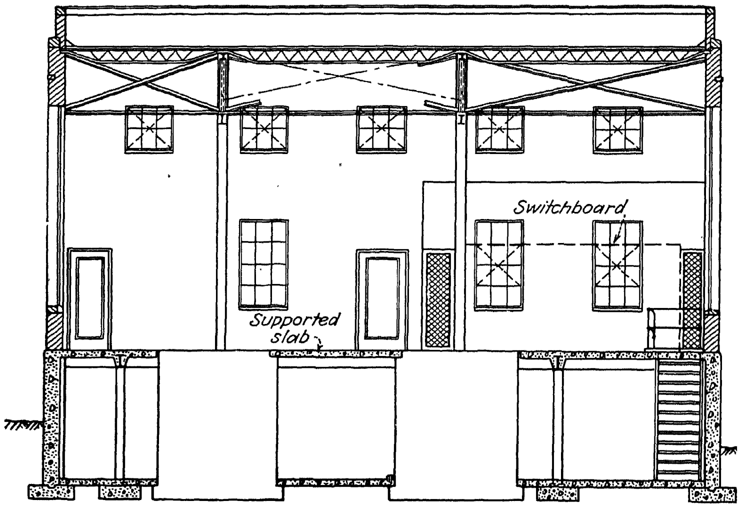


FIG. 44.—Plant designed with floor slab supported by the adjoining engine foundations.

transmission at that point. In some special cases, engine foundations have been satisfactorily constructed integral with the basement floor but isolated from the operating floor.

Operating-floor sections between engine foundations are often built as reinforced-concrete slabs resting on the adjoining engine foundations and not connected to the other portions of the operating floor forming part of the building proper. While this construction, shown in Fig. 44, is satisfactory, it is open to the objection that it is practically impossible to obtain good bearing for the slab throughout the length of the foundation block. This lack of satisfactory bearing may result in considerable local

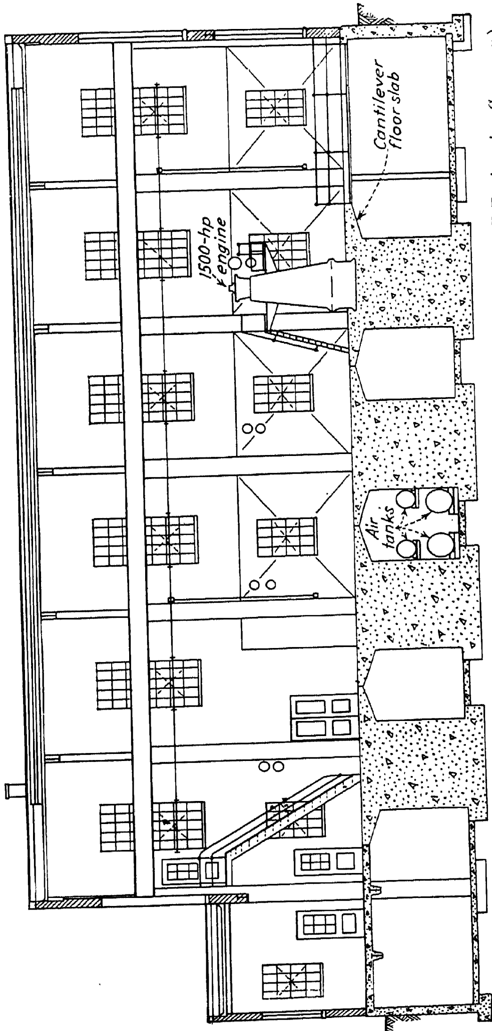


Fig. 45.—Floor slabs built as cantilevers from engine foundations. (*Courtesy of Burns & McDonnell Engineering Company.*)

vibration in the slab when the engine on either side of it is in operation. The vibration in such supported slabs is often greatly in excess of the vibration of the engine foundation on which it rests.

In order to overcome this rather serious objection to supported floor slabs, these sections between engine foundations have in several instances been designed and constructed as integral parts of the engine foundation. The foundation has constructed along each side a reinforced-concrete cantilever section extending out half the distance to the adjoining foundation as shown in Fig. 45. In this type of construction it is frankly recognized that the floor will vibrate with the engine, but the vibration in the floor section forming a part of the engine foundation has not been so noticeable as has been the vibration where the floor slab was constructed as shown in Fig. 44.

54. Fire Walls.—In the construction of a pipe-line pumping station handling gasoline, a fire wall is built in the station isolating the engines from the pumps. The engine drive shaft extends through the fire wall connecting to the pumps on the other side with special means employed to seal the opening around the drive shaft to prevent fire passing from the engine to the pump room.

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CHAPTER VIII

SELECTING THE ENGINE

The selection of an internal-combustion engine for a particular installation necessitates a thorough study of the requirements and limitations of the service as well as the characteristics and limitations of the engines available. In order that the most satisfactory engine may be selected for the particular service requirements contemplated, the advantages of the various types of construction available as well as their limitations should be known. There are many excellent engines in the market today. In most instances they have been developed and offered for sale by the manufacturers to meet definite commercial requirements. The purchaser's problem is to select that make and type of engine from the wide choice of excellent models available which best suits his conditions.

It is not possible in a single chapter to cover in detail the design of internal-combustion engines or even to describe the many excellent makes available. Rather than attempt to describe any particular engine designs, this chapter will deal in general terms with the matter of engine selection. A brief description of the operation of internal-combustion engines as well as explanations of terms dealing with the performance and description of engines is included for those not familiar with engine design and operation.

55. How Internal-combustion Engines Work.—An internal-combustion engine is one that derives its power from the burning of the fuel within the engine cylinder or an external chamber directly connected to the cylinder. This combustion produces increased temperature and pressure in the combustion space, and the developed gas pressure forces the piston out of the cylinder. This force exerted by the gas on the piston is transmitted through the connecting rod to the crankshaft, resulting in a turning effort. As the piston moves out of the cylinder, the gas pressure constantly decreases until the piston reaches the end of its travel in the cylinder.

This action is best illustrated by the series of diagrams for a diesel engine in Fig. 46. In *A* the piston is starting to compress a charge of air in the cylinder. The pressure shown on the diagram at (*a*) is atmospheric. As the piston moves upward into the cylinder, the pressure of the air increases as shown in *B*. By

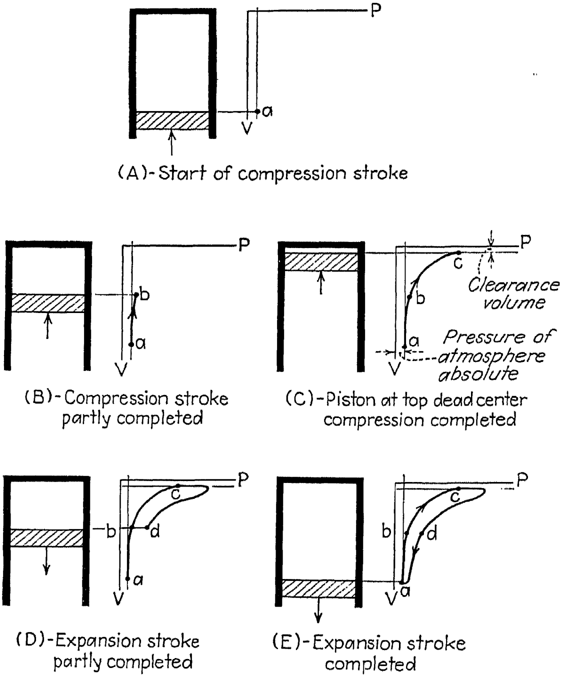


Fig. 46.—Diagrams showing compression and power strokes in a compression-ignition engine.

the time the piston has reached top dead center, as shown in *C*, the pressure has increased to (*c*). Ignition of the fuel charge, which had been introduced just before the piston had completed its upward travel, increases the pressure and forces the piston out of the cylinder. When the piston reaches its extreme outward position of travel, shown in *E*, the compression and power strokes have been completed. The amount of work done by the piston is proportional to the area enclosed by the pressure-stroke curve

shown in *E*. The foregoing describes the power-producing portion of either a two-stroke or four-stroke diesel cycle.

The production of power by an engine involves introduction of fuel and air into the cylinder, compression of the mixture or of the air alone, ignition of the fuel charge, expansion of the burning fuel and inert gas in the cylinder forcing the piston outward and

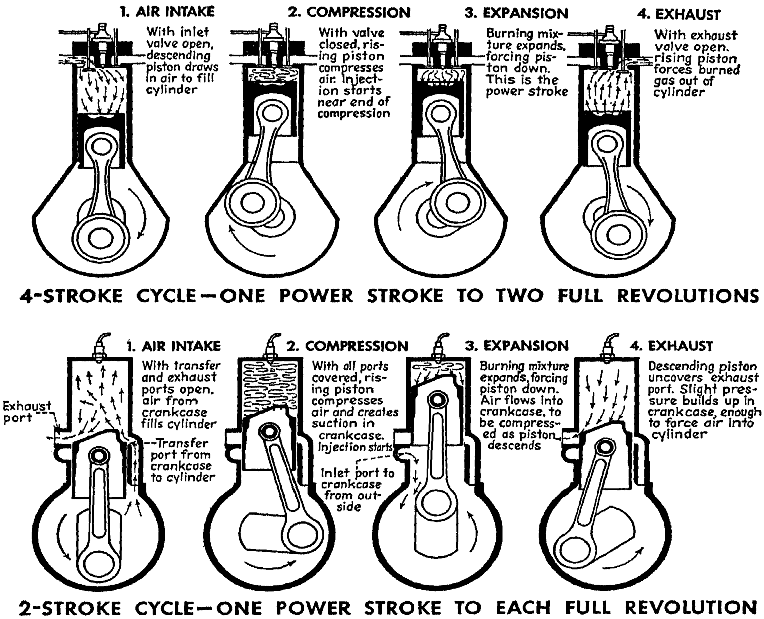
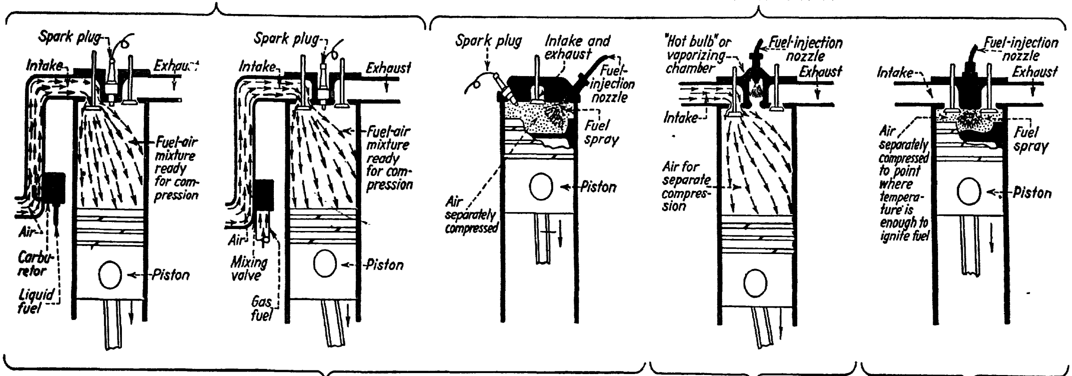


FIG. 47.—Diagrams illustrating operations in four-stroke- and two-stroke-cycle engines. (*Courtesy of Power.*)

producing power, and the ejection of the burned fuel charge and inert gas from the cylinder. This process is repeated in the cylinder each power cycle. If it requires but two piston strokes or a single revolution of the engine crankshaft for these events to occur, it is known as a *two-stroke cycle*, whereas if the events require four piston strokes or two complete crankshaft revolutions for completion, it is called a *four-stroke cycle*. Figure 47 shows the events of the four-stroke and two-stroke cycles diagrammatically.

FUEL AND AIR MIXED OUTSIDE CYLINDER, BEFORE COMPRESSION

AIR ONLY COMPRESSED IN CYLINDER; FUEL SPRAYED INTO CYLINDER WHERE MIXING OCCURS



SPARK IGNITION | **SURFACE IGNITION** | **COMPRESSION IGNITION**

GASOLINE ENGINE

1. Carburetor vaporizes fuel and mixes it with air in proper proportions.
2. Suction stroke, with intake valve open, fills cylinder with air-fuel mixture.
3. Compression stroke raises pressure of air-fuel mixture to 80-180 lb.
4. Spark ignites mixture near end of compression stroke.
5. The fired mixture expands, pushing piston down to produce "working" stroke.
6. Exhaust valve opens; rising piston clears cylinder of burned gas.

GAS ENGINE

1. Mixing valve blends air and fuel in proper proportions.
2. Suction stroke, with intake valve open, fills cylinder with air-fuel mixture.
3. Compression stroke raises mixture pressure to 70-300 lb, depending on fuel.
4. Spark ignites mixture near end of compression stroke.
5. The fired mixture expands, pushing piston down to produce "working" stroke.
6. Exhaust valve opens; rising piston clears cylinder of burned gas.

HESSELMAN ENGINE

1. Suction stroke, with air-intake valve open, fills cylinder with air.
2. Compression stroke raises air pressure to 115-135 lb.
3. Fuel injection is completed before compression stroke ends.
4. Spark (just before compression ends) ignites fuel vaporized by compression and stored heat.
5. Combustion expansion pushes down piston, producing "working" stroke.
6. Exhaust valve opens; rising piston clears cylinder of burned gas.

VAPORIZING OIL ENGINE

1. Suction stroke, with air-intake valve open, fills cylinder with air.
2. Pump injects fuel to vaporizer; vapor formed mixes with burned gas and doesn't burn.
3. Compression stroke raises pressure to 60-150 lb; forces air into vaporizer.
4. Heated surface of vaporizer chamber ignites fuel-air mixture near end of compression.
5. Combustion occurs with explosive force, producing "working" stroke.
6. Exhaust valve opens; rising piston clears cylinder of burned gas.

DIESEL ENGINE

1. Suction stroke, with air-intake valve open, fills cylinder with air.
2. Compression stroke raises air pressure to 350-600 lb.
3. Fuel injection starts at or near the end of the compression stroke.
4. High air temperatures, caused by compression, ignite fuel.
5. Combustion expansion pushes down piston, producing "working" stroke.
6. Exhaust valve opens; rising piston clears cylinder of burned gas.

FIG. 48.—Types of internal-combustion engines. (*Courtesy of Power.*)

There are several types of internal-combustion engines resulting from the nature of the fuel employed, the method of introducing the fuel into the cylinder, and the method employed for igniting the fuel charge. The more common types are shown in Fig. 48. To these must be added the gas-diesel, burning natural gas on the diesel principle.

56. Types of Internal-combustion Engines.—The classification of internal-combustion engines is somewhat involved owing largely to the many improvements and new developments brought forth in recent years. Unfortunately, some of the currently

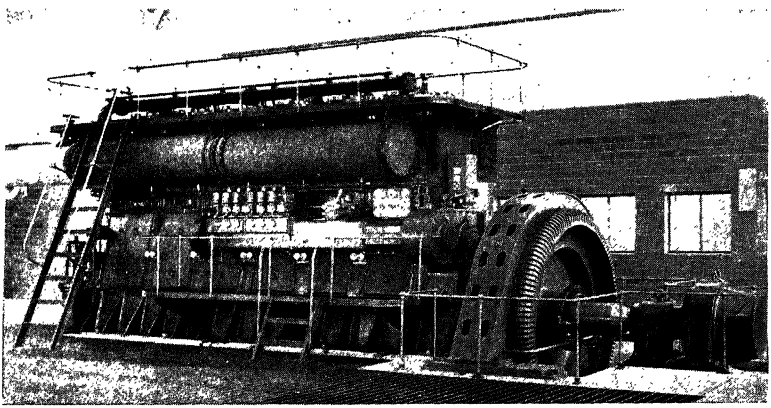


FIG. 49.—2,040-hp two-stroke-cycle diesel engine driving an alternator in a utility plant. (*Courtesy of Busch-Sulzer Bros. Diesel Engine Company.*)

accepted terms used to describe various types of engines are very broad. For example, the term *diesel engine* has grown through the years to cover many diversified kinds of engines practically none of which bear any resemblance to the engine conceived by Dr. Diesel. In fact, most people consider that any engine which does not use spark plugs to ignite the fuel charge is a diesel engine.

Any rational method for a basic classification of internal-combustion engines must take into consideration the kind of fuel burned, the method of introducing the fuel into the cylinder, the method for igniting the fuel charge, and the stroke-cycle employed. Service requirements and mechanical construction may also serve to classify an engine, but basically the type of engine is not determined by them. These factors are set forth in outline form as follows.

OUTLINE OF ENGINE CLASSIFICATION

I. Basic classification.
  A. Type fuel.
    1. Gasoline.
    2. Gas.
    3. Oil.
    4. Both gas and oil (convertible).
  B. Fuel admission.
    1. Carburetor (aspiration).
    2. Injection.
      a. Air blast.
      b. Mechanical.
    3. Compression (for gas).
  C. Fuel ignition.
    1. Heat of compression.
    2. Surface ignition.
    3. Spark ignition.
  D. Stroke cycle.
    1. Two strokes.
    2. Four strokes.
II. Modifying conditions.
  A. Service requirements.
    1. Stationary.
    2. Marine.
    3. Railway.
    4. Automotive.
    5. Aircraft.
    6. Power shovels, cranes, etc.
  B. Mechanical construction.
    1. Stroke action.
      a. Single acting.
      b. Double acting.
    2. Rotative speed.
      a. Slow.
      b. Medium.
      c. High.
    3. Cylinder arrangement.
      a. Vertical.
      b. Horizontal.
      c. Vee.
      d. Radial.
    4. Piston construction.
      a. Crosshead.
      b. Trunk.
    5. Piston cooling.
      a. Uncooled.
      b. Water cooled.
      c. Oil cooled.

57. Definition of Terms.—There are several terms used in conjunction with engine performance that should be clearly understood. Definition of these terms follows.

Displacement.—The displacement of an engine is the volume, expressed either in cubic feet or cubic inches per minute, which is swept by all the pistons during the power strokes and is equal to

$$V_d = L \times A \times n \tag{2}$$

where V_d = displacement volume per minute, cubic feet or cubic inches.
L = piston stroke, feet or inches.
A = piston area, square feet or square inches.
n = power strokes per minute for all cylinders.

Mean Indicated Pressure.—The mean indicated pressure (p_i) is the average net pressure in pounds per square inch acting on the piston during the power stroke only. It is determined by direct measurement through the use of an engine indicator.

Indicated Horsepower.—The indicated horsepower (P_i) of an engine is the total horsepower developed by all the engine cylinders without allowance for friction and other losses within the engine. In the case of slow-speed engines, it may be calculated by using values for mean indicated pressure obtained from indicator cards in the following equation:

$$P_i = \frac{p_i A L n}{33{,}000} \tag{3}$$

where P_i = indicated horsepower.
p_i = mean indicated pressure, pounds per square inch.
A = piston area, square inch.
L = piston stroke, feet.
n = total power strokes per minute.

The indicated horsepower for high-speed engines is obtained by adding to the brake horsepower of the engine the frictional horsepower determined by dynamometer motoring tests, or

$$P_i = P_m + P_f \tag{4}$$

where P_m = brake horsepower.
P_f = horsepower absorbed by friction.

Friction-horsepower determinations by means of dynamometer motoring tests are not extremely accurate. Consequently indi-

cated-horsepower values determined by Eq. (4) are influenced by the inaccuracy existing in the friction-horsepower determination.

Brake Horsepower.—The brake horsepower (P_a) of an engine is the horsepower delivered to the engine crankshaft. It can be determined by direct test either by the use of a prony brake or dynamometer (see Art. 251, Chap. XX).

Brake Mean Effective Pressure.—The brake mean effective pressure (p_e) of an engine cannot be measured and is a theoretical

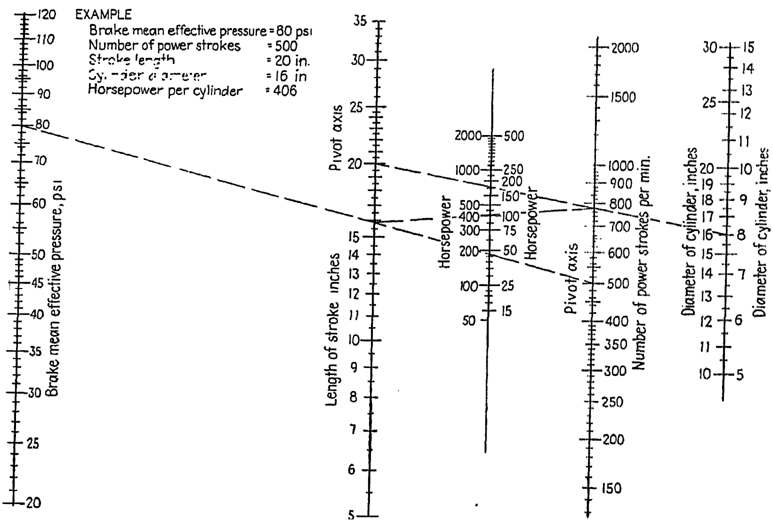


FIG. 50.—Nomographic chart for determining brake mean effective pressure.

value which is used for the comparison of mean pressures in engine cylinders. It is determined by calculation from brake-horsepower tests of an engine by use of the following equation:

$$\frac{33{,}000 \times P_a}{LAn} \tag{5}$$

Most operating men and designers speak of brake mean effective pressure as though it were an actuality and use this theoretical value as a convenient means for comparing various engine ratings. There is considerable justification for this procedure since the

brake mean effective pressure is readily calculated from the shaft horsepower and other known characteristics of the engine.

Mechanical Efficiency.—The mechanical efficiency (e_m) of an engine is the ratio of the brake horsepower to the indicated horsepower or the ratio of the brake mean effective pressure to the mean indicated pressure.

$$e_m = \frac{P_m}{P_i} \times 100 = \frac{p_e}{p_i} \times 100 \qquad \text{per cent} \tag{6}$$

Piston Speed.—The piston speed (V_p) of an engine is the total travel of the piston in a cylinder during 1 min.

$$V_p = 2Ln_o$$

where V_p = piston speed, feet per minute.
L = piston stroke, feet.
n_o = engine rpm.

Piston speeds for diesel engines in stationary service vary from 650 to 1,400 fpm, although the most commonly used piston speeds for large engines range from 1,000 to 1,200 fpm.

The nomographic chart, Fig. 51, provides a rapid means for determining piston speeds with the stroke ranging from 5 to 30 in. and rotative speeds from 125 to 2,000 rpm. A straightedge placed across the chart connecting the known stroke and rpm will cross the value for piston speed on the center scale. Thus for an engine with a 28-in. stroke operating at 225 rpm, the piston speed is 1,050 fpm, while an engine with an 8-in. stroke operating at 1,200 rpm has a piston speed of 1,600 fpm.

Compression Ratio.—The compression ratio of an internal-combustion engine is the ratio of the cylinder volume at the start of the compression stroke to the clearance volume above the piston when it has reached top dead center. Stated another way, it is the volume of air in the cylinder at the start of the compression stroke divided by the volume attained by this air at the end of the compression stroke.

58. Engine Fuel.—Fuels used in internal-combustion engines are usually hydrocarbons which may be either liquid or gas at ordinary temperatures and pressures. Liquid petroleum products including gasoline, kerosene, and fuel oils are used in the largest proportion of engines for fuel, although many engines utilize natural gas, sewage gas, manufactured gas, and blast-

furnace gas as fuel. Engines considered in this book burn fuel oil and gas. Separate chapters are devoted to consideration of liquid and gaseous fuels.

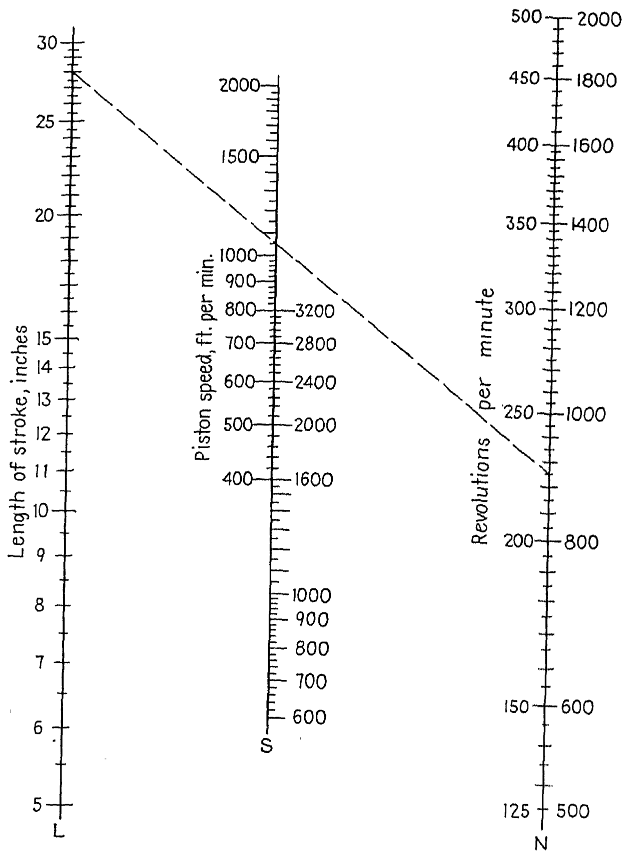


FIG. 51.—Nomographic chart for determining piston speed.

59. Fuel Admission.—There are three basic methods for introducing fuel into an engine cylinder. These are induction, injection, and compression.

Induction of fuel into an engine cylinder is familiar to those conversant with the operation of an automobile engine. The fuel is vaporized in a caburetor or other suitable device for mixing the fuel with the air drawn into the cylinder during the suction stroke. The engine then compresses a combustible mixture of fuel and air. Since the compression of any gas or mixture of gases raises their temperature, it is necessary that the pressure increase through compression be kept below the ignition temperature of the fuel-air mixture; otherwise ignition of the mixture would occur before such ignition was desired. Whenever an engine is equipped for induction of the fuel with the air, it is customary to use an electric spark for igniting the fuel charge.

A mixing valve is used on a gas engine for proportioning the combustible gas and air drawn into the cylinder. The gas and air entering the cylinder must be proportioned and mixed correctly to ensure proper combustion. The load on an engine varies, and it is necessary that the proper amount of fuel be supplied for the load carried by the engine. Thus the mixing valve must be capable of maintaining the correct proportioning of gas and air for all rates at which fuel is demanded by the engine.

There are three types of mixing valves in general use. These are the mechanical, vacuum, and Venturi types.

The *mechanical type*, utilizing throttling to regulate the quantity of gas flowing, is employed on the Worthington gas engine, Fig. 52. It is provided with separately adjustable valves for controlling air and gas.

Air is controlled by a double-seated valve, while the gas is regulated by a single long conical valve. Each valve is adjusted separately to obtain the proper fuel-air ratio. After this adjustment has been made for a particular gas, this ratio is maintained constant regardless of the load on the engine.

In the *vacuum-type mixer*, a piston valve, which controls the rate of flow of gas and air through orifices, is actuated by the vacuum produced by the engine on the suction stroke. The quantity of mixture is controlled by a governor-actuated butterfly valve. Mixers of this type usually require the gas be delivered to the mixer at 4 to 8 in. of water pressure.

In the *Venturi type*, gas and air are mixed in the throat of the Venturi tube by the vacuum created during the suction stroke. Gas and air are at atmospheric pressure in the Venturi throat.

After the gas has been adjusted for the proper fuel-air ratio it will remain constant at all loads. The quantity of air-gas mixture admitted to the engine cylinders is controlled by a governor-actuated butterfly valve.

Injection of fuel into an engine cylinder is usually done just before the maximum pressure occurs on the compression stroke,

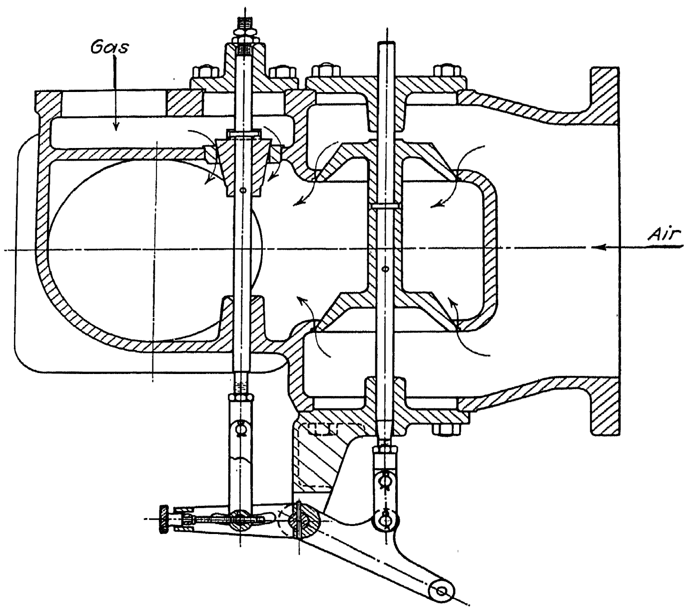
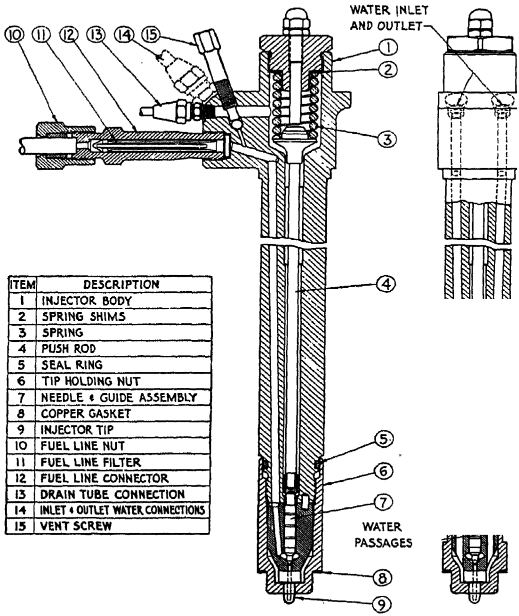


FIG. 52.—Section through gas mixing valve used on Worthington gas engine. (*Courtesy of Worthington Pump and Machinery Corporation.*)

in other words, just prior to the piston reaching top dead center. As a result, it is necessary to force the fuel charge into the cylinder against considerable pressure, usually from 500 to 700 psi. In order to get the fuel into the cylinder, it can be forced in by means of a high-pressure air blast, usually about 1,200 psi, or it can be injected through the use of a suitable high-pressure pump. The former method of fuel introduction is known as *air injection*, while the latter is called *mechanical injection*. Forced injection of fuel is employed on all diesel engines.

When air injection of the fuel is employed, it is necessary to have a suitable high-pressure compressor to supply the necessary injection air. This compressor is usually built into the engine.

The earlier diesel engines employed air exclusively for introducing the fuel charge into the cylinder, primarily because mechanical skill and precision were not sufficiently advanced to permit con-



ITEM	DESCRIPTION
1	INJECTOR BODY
2	SPRING SHIMS
3	SPRING
4	PUSH ROD
5	SEAL RING
6	TIP HOLDING NUT
7	NEEDLE & GUIDE ASSEMBLY
8	COPPER GASKET
9	INJECTOR TIP
10	FUEL LINE NUT
11	FUEL LINE FILTER
12	FUEL LINE CONNECTOR
13	DRAIN TUBE CONNECTION
14	INLET & OUTLET WATER CONNECTIONS
15	VENT SCREW

FIG. 53.—Mechanical-injection nozzle equipped for water cooling. (*Courtesy of Aircraft & Diesel Equipment Corporation.*)

struction of suitable mechanical-injection accessories for proper oil-injection and load-distribution requirements. Improvements in the art of constructing fuel-injection accessories have overcome these earlier mechanical defects. Today the majority of engines in use or being built are equipped with the mechanical-injection system.

Cost and efficiency are primarily the reasons for this pronounced trend to mechanical fuel injection on diesel engines.

It costs less to build the mechanical-injection system required for an engine than it does to build the high-pressure air compressor and other component parts of an adequate air-injection system. Power is required to drive the compressor, and this power, ranging from 8 to 10 per cent of the capacity of the engine at full-load rating, must be added to the parasite power losses. Some portion of this power is recovered upon expansion of this

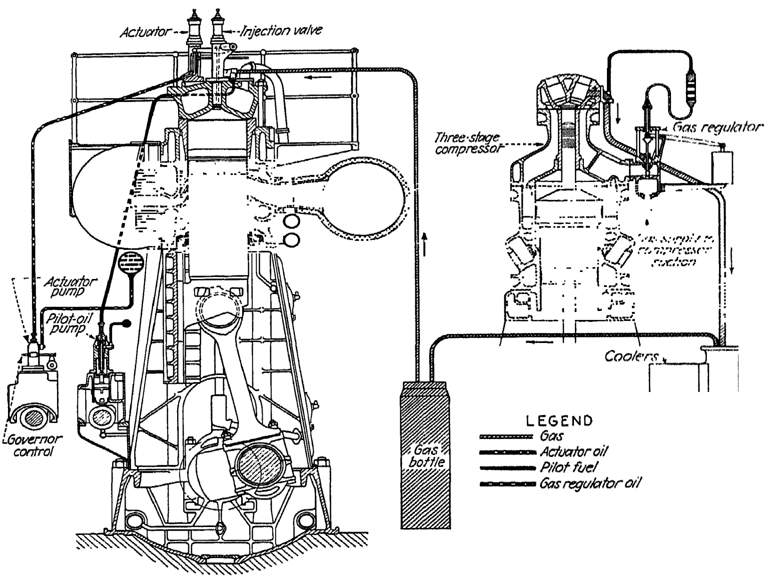


Fig. 54.—Schematic arrangement of gas details and piping of Nordberg convertible gas-burning diesel. (*Courtesy of Nordberg Manufacturing Company.*)

air in the engine cylinders. The mechanical-injection engine is inherently the more efficient type owing to the absence of the relatively large power demand of the compressor required with the air-injection system.

It is only when the heavier fuel oils are used in an engine that the type of fuel-injection system employed in a particular construction becomes of prime importance. Until recent years, only air-injection equipment was considered capable of handling the heavy usable residual fuel oils. Developments during the

past decade, however, have brought forth many important improvements in mechanical-injection equipment. Today both mechanical and air injection are successfully employed for handling heavy fuel oils.

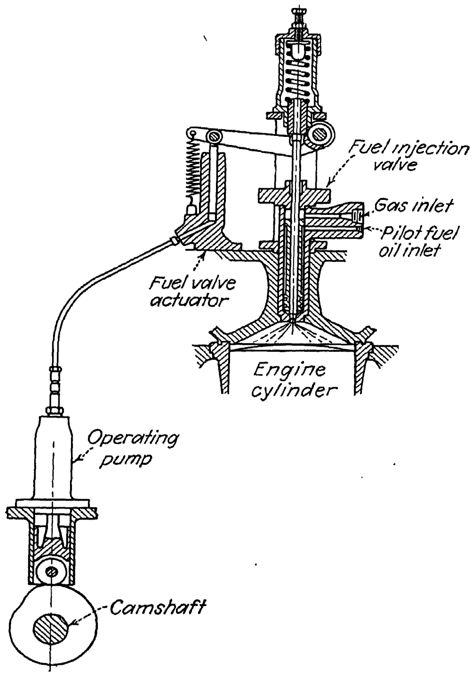


FIG. 55.—Hydraulic valve operator employed on Nordberg convertible gas-burning diesel. (*Courtesy of Nordberg Manufacturing Company.*)

Compression of the fuel charge as a means of introducing it into the engine cylinder is employed in the Nordberg gas-diesel engine, Fig. 54. This engine compresses the gas in a three-stage compressor to 1,100 psi before injecting it into the cylinder. Simultaneously with the injection of the high-pressure gas, a small quantity of fuel oil is also introduced by means of a mechanical-injection pump. This oil is used to stabilize combustion of the gas and ensure that each charge of gas entering the cylinder

will be ignited. The fuel valve is hydraulically operated as shown in Fig. 55.

Tests of the Nordberg gas-diesel show that approximately 6 per cent of the heat required for producing power is supplied by the pilot oil, the remaining fuel being gas. The initial engines of this type delivered a constant quantity of pilot oil regardless of load on the engine, although the present practice is to vary the amount of pilot oil injected with the fuel charge in proportion to the load on the engine.

The National Gas and Oil Engine Company, Ltd., of England has developed a dual-fuel engine known as the *oil-cum-gas engine*. Tests on a 200-hp four-stroke-cycle, four-cylinder, 11- by 15-in., 428-rpm unit built by this company showed that at full load the total heat consumption was 7,659 Btu per bhp-hr. Of this heat, 90 per cent was supplied by gas having a high-heat value of 482 Btu per cu ft, and the remaining 10 per cent by fuel oil. A compression ratio of 13.7 is used, and the governor, in addition to varying the quantity of gas admitted for the load carried, also adjusts for the variation in heating value of the fuel, thereby maintaining a constant quality of the gas-air mixture.

60. Fuel Ignition.—After the fuel charge has been introduced into the engine cylinder, it is necessary that it be ignited. There are three methods employed for performing this function.

Spark ignition is used in all engines burning gasoline or gas where the fuel charge enters the cylinder, with the air for supporting combustion, through a carburetor or gas-mixing valve. Spark ignition is also used in the Hesselman engine. There are two systems of spark ignition in general use today. The first employs an electric battery and high-tension coil, while the second uses a high-tension magneto. A suitable distributor for firing the spark plugs in proper sequence is used in either case.

Surface ignition employs a hot tube, bulb, or plate usually incorporated in the cylinder head for igniting the fuel charge. The surface-ignition method had its inception in the low-pressure oil engines developed some 40 years ago, where the air charge in the cylinder was first compressed to a relatively low pressure before oil was injected into the cylinder. Since the temperature of the compressed air within the cylinder was not high enough to ignite the fuel charge upon introduction, it was necessary to provide a means for heating a portion of the cylinder head above the

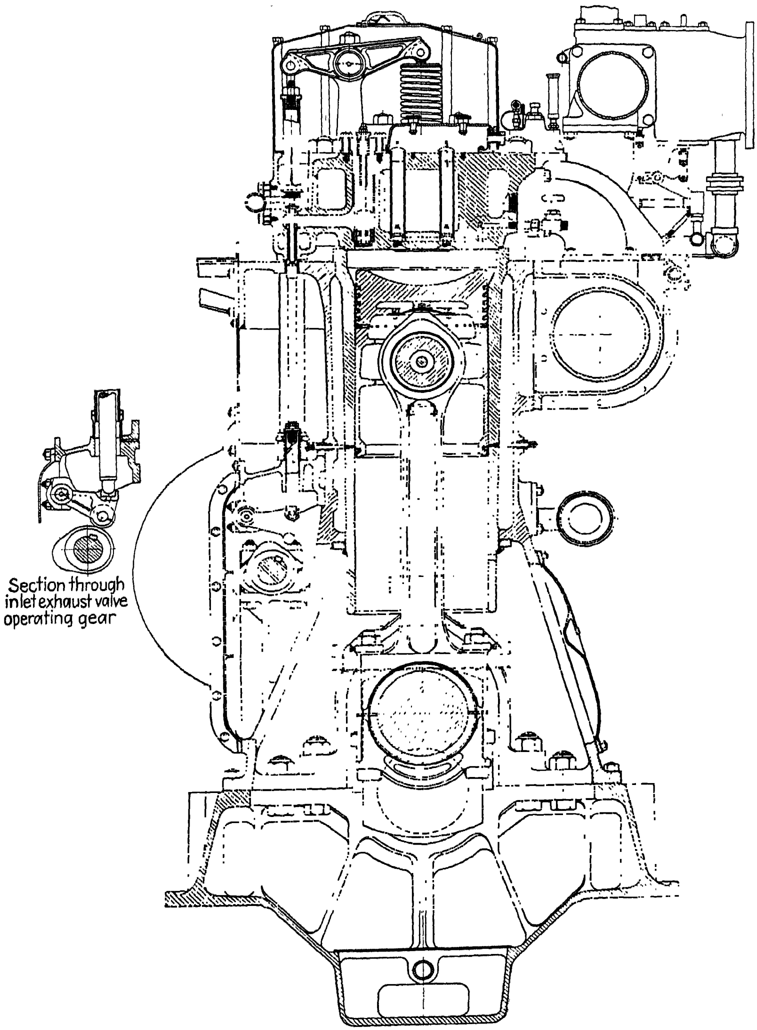


Fig. 56.—Sectional view of Worthington Type EEG gas engine. (*Courtesy of Worthington Pump and Machinery Corporation.*)

ignition temperature of the oil before starting the engine. After the engine began operating, this uncooled igniting surface or tube was maintained at a sufficiently high temperature by the combustion of fuel in the cylinder to ensure ignition of each succeeding fuel charge.

Compression ignition was first proposed by Dr. Diesel in 1893. He advanced the idea that if air were compressed to a sufficient

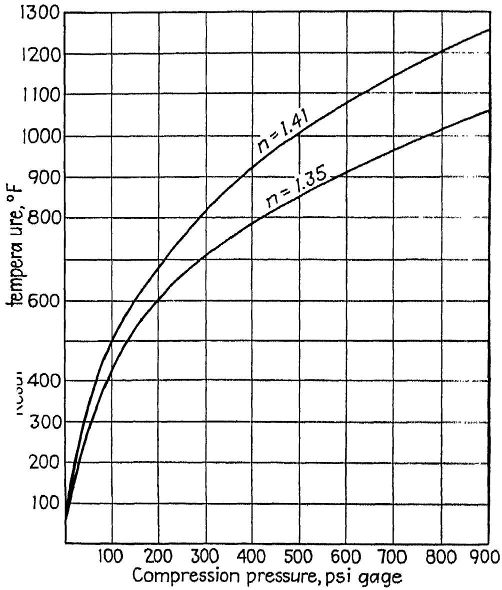


FIG. 57.—Chart showing theoretical variation in air temperature in relation to pressure to which it is compressed.

pressure in an insulated cylinder it would attain a temperature high enough to ignite the fuel charge when introduced into the cylinder. This principle, modified to the extent that the cylinder is cooled instead of being insulated, is used today in all compression-ignition engines. The air is first compressed to a value somewhere between 450 and 700 psi, depending upon the design of the engine. The temperature attained by the air upon compression is sufficient to ignite the fuel charge when introduced into the cylinder.

Theoretical studies covering the compression of so-called perfect gases with the ratio of the specific heat at constant pressure and constant volume maintained constant, and the compression following a theoretical adiabatic cycle, will produce pressure-temperature relationships as shown for $n = 1.41$ in Fig. 57. In the actual compression of air in a cylinder, the value used for n in the equation

$$T_2 = T_1 \left(\frac{p_2}{p_1}\right)^{1-\frac{1}{n}}$$

may vary considerably from the value of 1.41. As its value decreases, the temperature attained upon compression of the air is decreased as shown by the curve for $n = 1.35$.

In reality, the ratio of the specific heats at constant pressure and constant temperature is continually varying during compression. Consequently the only means for determining the actual temperature attained by the air charge in the cylinder is from actual experiment. The diameter of the cylinder, cylinder-cooling methods, temperature and pressure of the incoming air all influence the final temperature attained by the air charge.[1]

61. Stroke Cycles.—Reference has already been made to the fact that engines operate on either the two-stroke or four-stroke cycle.

Two-stroke-cycle engines are those in which combustion occurs in each cylinder once every revolution. Engines of the single-acting type have been built in sizes up to 750 hp per cylinder, and by the use of double-acting units firing on both sides of the piston alternately, the power developed per cylinder has been increased to well over 2,000 hp per cylinder. The largest diesel engine so far constructed is the Burmister and Wain unit of 22,500 hp in eight cylinders installed in an electric generating station at Copenhagen, Denmark. This unit is of the double-acting two-stroke-cycle design. Because of the absence of valves in the

[1] For those who desire to consider the matter of theory dealing with temperatures attained upon compression, the following works should be consulted:

LICHTY, L. C., "Internal Combustion Engines," 5th ed., McGraw-Hill Book Company, Inc., New York, 1939.

SHEPHERD, HAROLD F., "Diesel Engine Design," John Wiley & Sons, Inc., New York, 1935.

cylinder heads of most two-stroke-engine designs, it is contended that a much more symmetrical and therefore more rugged head can be produced than is possible in the case of the four-stroke-cycle engines, where either two or four valves are required in the cylinder head. It is further claimed that this absence of valves results in an engine with fewer moving parts over the cylinder heads.

It is necessary to provide some means for scavenging the cylinders of a two-stroke-cycle engine so that sufficient fresh air is present for sustaining combustion of the fuel during the firing portion of the power stroke. This scavenging can be accomplished in several ways, such as by crankcase scavenging (Fig. 47), a piston-type pump mounted on the engine, direct-driven blower, or a blower driven from a separate power source.

Brake mean effective pressures on the basis of present-day two-stroke-cycle engine ratings range from 33 to 68 psi and higher.

Four-stroke-cycle engines are those in which combustion occurs in each cylinder once every second revolution. The tendency is for the four-stroke-cycle engine to predominate in the smaller cylinder sizes of unsupercharged units ranging from approximately 200 hp per cylinder down. For cylinder capacities of approximately 200 hp and greater, the two-stroke-cycle engine is cheaper to build. When individual cylinder capacities exceed approximately 500 hp, it is necessary to employ the two-stroke cycle.

There appears to be a tendency among those engine builders in the United States manufacturing both four-stroke- and two-stroke-cycle engines to limit the size of their four-stroke-cycle engines to approximately 150 hp per cylinder, while their two-stroke-cycle units are built for capacities exceeding 100 hp per cylinder.

Brake mean effective pressures on the basis of present-day four-stroke-cycle nonsupercharged engine ratings range from about 60 to 100 psi, and for supercharged engines in the approximate range of 100 to 130 psi.

62. Mechanical Construction.—There are certain mechanical design features which classify an engine, and some of these are presented here. It should be realized that only those general features are treated which serve to classify the general mechanical construction of the engine.

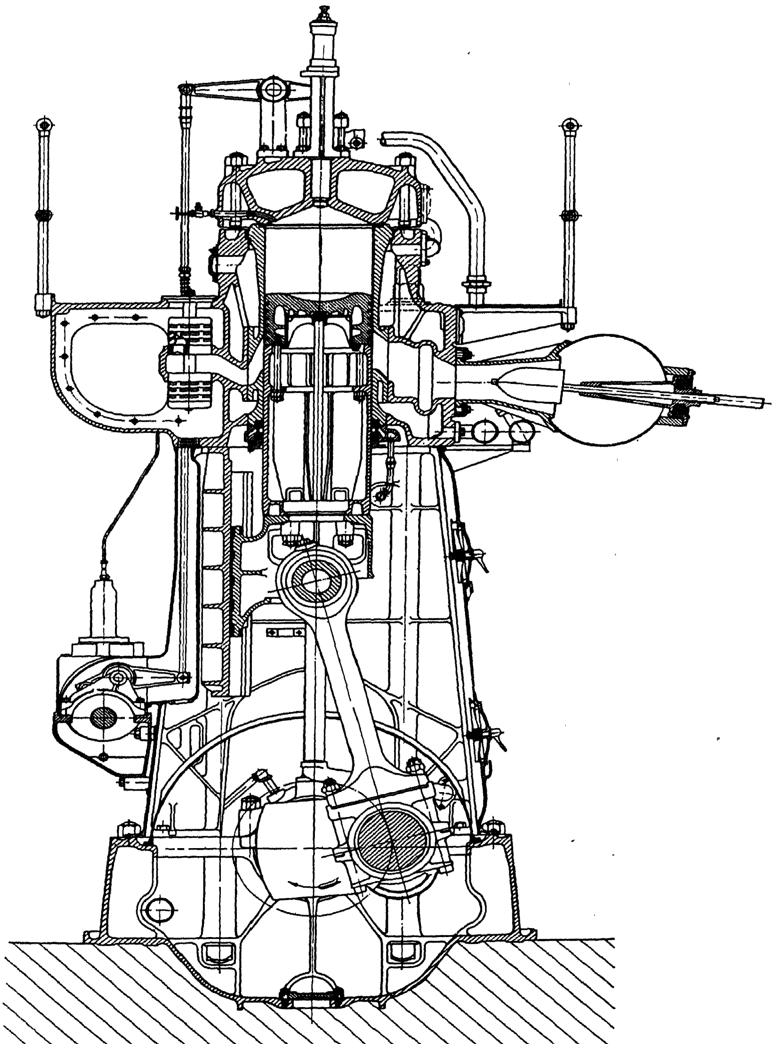


FIG. 58.—Sectional view of engine with cross-head piston construction. (*Courtesy of Nordberg Manufacturing Company.*)

Engines may be designed to develop power on only one side of the piston (*single acting*), or they may be designed to develop power on both sides of the piston (*double acting*). The greater portion of the internal-combustion engines built in the United States are of the single-acting type. In general, double-acting engines employ the two-stroke cycle and are built in sizes from about 100 to over 2,000 hp per cylinder. Double-acting four-stroke-cycle engines, because of mechanical complications involved in the use of two sets of exhaust and intake valves for each end of each cylinder, are now generally considered commercially impractical.

Cylinder arrangements for internal-combustion engines may be either horizontal, vertical, vee type, or radial. Where the large slow-speed engines are used, the vertical type has predominated for many years. Recently, many vee-type engines have been installed as prime movers for electrical generating units in industrial establishments, office buildings, and other uses such as for railway locomotives. Many horizontal engines are used for oil pumping, gas compression, and for pipe-line service, particularly for natural-gas pumping. Most horizontal engines use gas for fuel, have two, three, or four cylinders arranged either singly or in tandem, single or double acting.

Piston construction is usually described as being of the crosshead or trunk type, Figs. 58 and 59. Crosshead-type pistons, as the name implies, employ a crosshead in connection with the piston which serves both as a guide and as a medium for absorbing side thrust resulting from the angularity of the connecting rod. There are numerous designs of crossheads in use, although the slipper type predominates. This type of crosshead, originally developed for marine engines, has been adapted for the larger stationary-type engines. The trunk-type piston has no separate crosshead, the side thrust being absorbed by the relatively long piston skirt. As pointed out by Purday,[1] there appears to be no limit to the size or power of a trunk piston. Practically all engines built in the United States rate 300 hp per cylinder or less employ trunk-type pistons. Engines having capacities exceeding 650 hp per cylinder equipped with trunk pistons are in successful operation.

[1] PURDAY, H. F. P., "Diesel Engine Design," 4th ed., p. 309, D. Van Nostrand Company, Inc., New York, 1937.

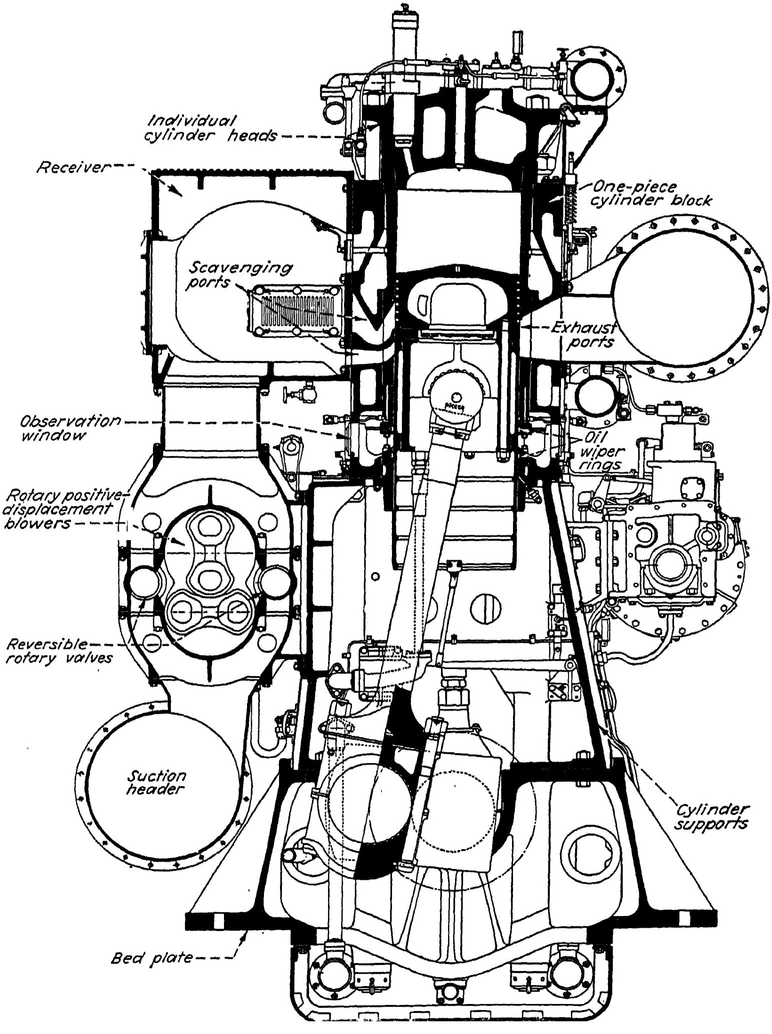


Fig. 59.—Sectional view of Busch-Sulzer Bros. engine with trunk piston construction. (*Courtesy of Power.*)

Piston cooling is employed on practically all the large, slow-speed engines. Cooling is accomplished by circulating water or oil through the piston head. The tendency, apparently, is to favor oil cooling of pistons in preference to water cooling owing largely to the fact that a leak in the water-cooling system of a piston would contaminate the lubricating-oil supply for the main bearings. In most engines with liquid-cooled pistons, the piston head is provided with some type of labyrinth to ensure that the cooling medium reaches all parts of the head. Many internal-

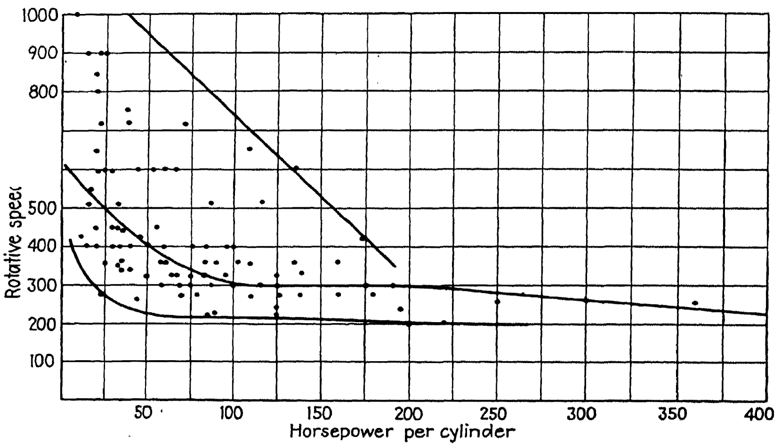


FIG. 60.—Relationship between horsepower per cylinder and rotative for modern stationary diesel engines.

combustion engines do not cool the piston, since the temperatures attained by the metal in the crown and upper walls are not high enough to cause damage to the piston. The decision to cool or not cool the pistons in a particular type of engine is a matter of design which is influenced by the diameter of the piston, material used for the piston, rate of heat transmission to and from the piston, and many other factors.

Rotative speeds for internal-combustion engines vary widely, as shown by Fig. 60. The data presented are for stationary engines only. In recent years there has been a distinct tendency to increase rotative speeds as new materials and improved processes of engine construction have become available. Since

the space occupied by engines of equal horsepower capacity is reduced with increase in rotative speed, considerable study as well as development has been devoted to the design and operation of engines with higher and higher rotative speeds. Space limitations in railway service, for example, necessitate the use of high-speed engines in order to provide the horsepower required within the space limitations imposed. In central-station service, the tendency has been to employ slow-speed units, operating at 400 rpm and less, in a majority of instances, although there are instances where engines operating at 600 and 900 rpm are being used for this type of service.

Judge[1] points out that the high-speed engine of the compression-ignition type has proved its suitability for small marine engines, stationary engines, aircraft engines, and for commercial vehicles, tractors, road rollers, excavators, railroad locomotives, and portable power plants.

63. Engine Cooling.—While Dr. Diesel's original conception of the compression-ignition engine involved an insulated cylinder, it was soon apparent that the temperatures resulting from the combustion of the fuel in such a cylinder would play havoc with the metals used for the cylinder and piston. As a consequence, it is necessary to provide cylinder cooling to limit the temperatures attained by the metal of which it is fabricated. The cooling method employed for an engine is influenced by the quantity of heat released per unit of cylinder volume, the diameter of the cylinder, and the time available for removal of that portion of the heat not used in the production of power. Cooling may consist of limiting the temperature of the cylinder walls and head only, although in some engines cooling of the pistons is also necessary. The cooling system in any engine must be simple and reliable.

Unfortunately, the design of the cooling-water system on some engines imposes excessive head losses when any appreciable amount of cooling water is circulated through the system. In general, however, engine designers have recognized the advantages of a vigorous water flow through the cylinder jackets as an aid to adequate cooling, and they have designed the water passages sufficiently large to permit a substantial water flow with a minimum head loss.

[1] JUDGE, ARTHUR W., "High Speed Diesel Engines," 3d ed., p. 10, D. Van Nostrand Company, Inc., New York, 1939.

64. Engine Efficiency.—The internal-combustion engine is essentially a heat-converting machine in which a portion of the heat resulting from the combustion of the fuel charge in the cylinder is converted into mechanical work. The efficiency of the conversion process is readily obtained when the heat input and power output are known. For an engine burning a liquid fuel, the thermal efficiency referred to the net brake-horsepower output is given by the equation

$$e_b = \frac{2{,}545}{q_1} \times 100 \qquad (7)$$

where e_b = thermal efficiency of engine.
q_1 = heat input of liquid fuel per brake horsepower-hour, high heat value.

When burning gaseous fuel, the thermal efficiency of the engine referred to the net brake-horsepower output is given by the equation

$$e_b = \frac{2{,}545}{q_c} \times 100 \qquad (8)$$

where q_c = heat input of gaseous fuel per brake horsepower-hour, high heat value.

It is often desirable to obtain the over-all efficiency of an engine-generator unit where an internal-combustion engine is driving an electrical generator. In this case, it is necessary to know the electrical output of the generator and the heat input to the engine to compute the over-all efficiency of the unit. For an engine burning liquid fuel, the over-all thermal efficiency of the assembly is determined by means of the following equation:

$$\times 100 \qquad (9)$$

where e_k = thermal efficiency of engine-generator assembly.
q_n = heat input of liquid fuel per kilowatt-hour, high heat value.

When the engine is burning a gaseous fuel, the over-all thermal efficiency of the engine-generator unit is computed by the use of the following equation:

$$q_f$$

where q_f = heat input of gaseous fuel per kilowatt-hour, high heat value.

Efficiency calculations referred to here are based upon the high heat value of the fuel used. There has been some objection to this method of efficiency determination on the part of gas-engine builders in the United States. Practice, however, has favored the use of the high heat value of fuels for all efficiency determinations made in the United States, because of the fact that the high heat value of the fuel is more readily and accurately determined than is the low heating value. A discussion of this matter in Chap. X shows that the low heating value of the combustible portions of gaseous fuels varies from 90.12 to 92.37 per cent of the high heat value, which is comparable to the variation for liquid fuels. The heating value of gaseous fuels is influenced primarily by the percentage of noncombustible gas present and not by the variation between the high and low heating values of the combustible gases available.

Theoretical studies of ideal engines indicate that for the same compression ratio those operating on the so-called *Otto* cycle are more efficient than those operating on the theoretical diesel cycle. In practical application, however, all engines with comparable compression ratios have substantially the same thermal efficiency. Parasite losses resulting from the power demands of auxiliaries, such as radiator fans, pumps, blowers, or compressors required for practical reasons, may influence this efficiency.

65. Engine Economy.—It is usual to express the performance of an internal-combustion engine burning liquid fuel in terms of the weight of fuel consumed per brake horsepower produced at one-half, three-quarters, and full load. The fact that the engine is a heat-converting machine is often lost sight of by doing this. In giving gas-engine performance, however, it is usual to express the fuel requirements per brake horsepower produced in terms of heat units (Btu).

Since both oil- and gas-burning engines are heat-converting engines, it would appear logical to express the fuel requirement in terms of Btu per brake horsepower-hour required for the various engine loads regardless of whether the engine used oil or gas fuel. Thus, instead of stating the fuel consumption of a diesel engine at full load as 0.4 lb of oil (heating value 19,000 Btu per lb) per horsepower-hour, the heat consumption would be

given as 7,600 Btu per bhp-hr at full load. At present, all fuel-consumption figures for diesels are given in pounds of fuel referred to a reference standard heat value. This really amounts to stating the performance in terms of heat units, although it is a cumbersome way of arriving at the desired result.

66. Engine Capacity.—The capacity of any cylinder to produce power is determined by the quantity of fuel that can be burned efficiently in that cylinder during each power stroke without creating excessive temperatures in the piston, cylinder walls, or cylinder head. One means for checking this maximum fuel-combustion rate is by conducting tests to determine the engine rating at which smoke first appears in the exhaust. A smoky exhaust indicates incomplete combustion of the fuel and hence decreased efficiency in the use of the heat in the fuel.

It is just as important that sufficient air be available to permit complete combustion of the fuel charge as it is to have sufficient fuel in the cylinder. If insufficient air is provided, incomplete combustion of the fuel results. In view of this condition, it is not surprising to find that it is impossible in some instances for the engine to produce its full load rating because excessive pressure drops in the air-intake and exhaust lines rob the engine cylinders of part of the air needed for complete combustion of the fuel charge. Two-stroke-cycle engines are particularly sensitive to extreme pressure losses in air-intake and exhaust systems, but four-stroke-cycle engines may also be adversely affected for the same reason.

Doolittle[1] made rather extensive investigations concerning the effect of the variation of air-intake pressure, exhaust back pressure, air temperature, and relative humidity on engine performance. His investigations indicated that for two-stroke-cycle engines the decrease in the air pressure to the engine and the increase in the exhaust back pressure were the elements having the greatest effect upon the capacity and fuel economy of a diesel engine. The results of these investigations dealing with air intake and exhaust are shown in Figs. 61 and 62.

The results obtained by Doolittle with small engines in the laboratory have been substantiated by the author in practical

[1] Doolittle, J. S., Effect of Variation in Atmospheric Conditions on Diesel Engine Performance, *Trans. A.S.M.E.*, vol. 63, No. 2, p. 91, February, 1941.

power-plant installations involving large slow-speed engines. In one instance a 1,000-hp 225-rpm four-stroke-cycle engine was moved from an existing power plant to a new installation. In moving the engine, both the air-intake and exhaust lines were shortened considerably and the sizes of both lines more than doubled in area. It was possible to get the engine to deliver only approximately 850 hp in its former installation. After the

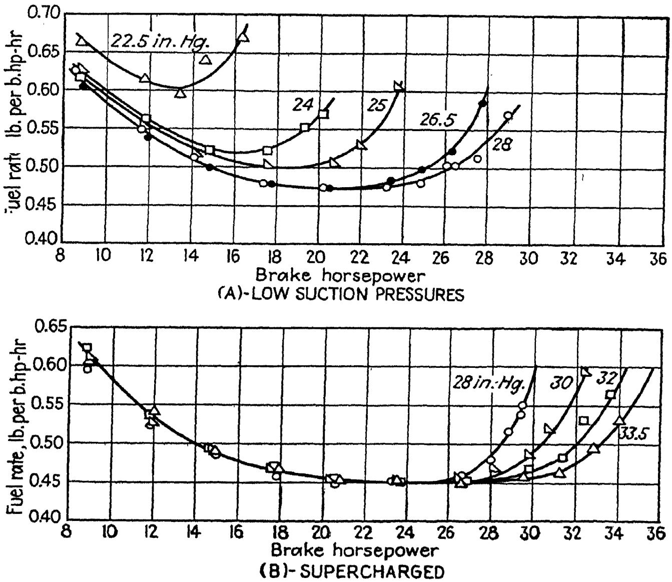


FIG. 61.—Effect of variation in intake pressure on two-stroke diesel-engine performance at atmospheric-exhaust pressure. (*Doolittle.*)

relocation, the engine carried a load of 1,000 hp with lower exhaust temperatures than were experienced at loads of 850 hp in its former location. In the second instance, an engine of approximately 3,000 hp was being tested. The pressure drop through the air-intake system to the engine was 0.77 in. Hg, and the exhaust pressure was 0.97 in. Hg. By a change in the air-intake system, reducing the pressure drop of the air to 0.07 in. Hg, the fuel economy of the engine at full load was improved by 2 per cent and the exhaust temperature was reduced 50 F. When the engine

was overloaded 10 per cent, the reduction in the inlet air pressure improved the fuel economy 6.5 per cent and reduced the exhaust temperature 115 F. The exhaust temperatures and fuel economies at three-quarters and one-half loads were not affected appreciably by the reduction in the intake air pressure. This

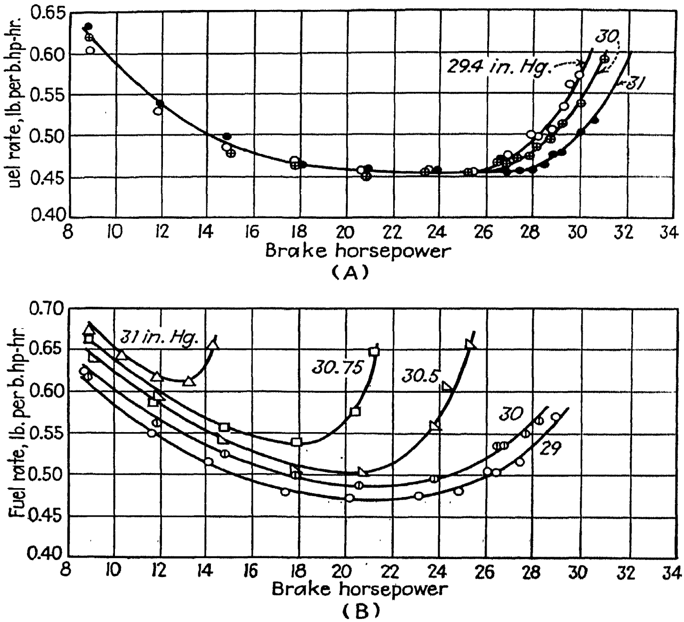


FIG. 62.—(*A*) Effect of variation in intake pressure on two-stroke-cycle engine performance with exhaust pressure equal to intake pressure. (*Doolittle.*) (*B*) Variation in exhaust pressure on two-stroke-cycle engine performance with a constant intake pressure of 28 in. Hg. (*Doolittle.*)

confirms the experiments made by Doolittle. It is thus seen that the correct design of the intake-air and exhaust systems for an internal-combustion engine may have a marked influence upon the performance of the engine.

In buying an engine, it is often necessary to determine just how conservatively it is rated, particularly when the unit is to be operated at full load for extended periods. By plotting exhaust temperatures against load as well as fuel consumption per unit of

output against load, from the guarantee data submitted by the manufacturer, it is often possible to detect whether or not an engine has been excessively rated. In general, the best fuel economy of any engine will be found to occur between three-quarters and full load. If the fuel-consumption curve tends to climb rather steeply between three-quarters and full load, it is very probable that the correct rating for the engine should be somewhere in the neighborhood of its three-quarters rating. This is illustrated by Fig. 63 in which curve *A* shows a conservatively rated engine, while curve *B* shows an engine that has been optimistically rated. The engine shown by curve *A* will, in all

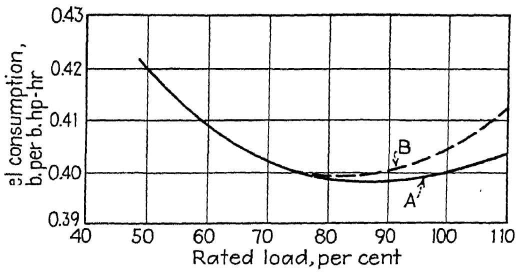


FIG. 63.—Comparison of fuel consumption for two proposed diesel engines.

probability, give better service and certainly more economical service at the loads above three-quarters load.

67. Brake-horsepower Rating.—At the present time there are several standards by which the capacity of an internal-combustion engine may be expressed. In the rating method that is rapidly being adopted for diesel engines, they are considered as being divided into two classifications with a distinct and separate method of capacity rating for each. The first group consists of engines running at rotative speeds below approximately 750 rpm, while the second group includes those engines operating at higher rotative speeds.

For engines operating at less than about 750 rpm, the currently accepted commercial or standard sea-level rating is the net brake-horsepower output developed *continuously* at the crankshaft coupling or power take-off end of an engine in good operating condition located at an altitude less than 1,500 ft above sea level with the atmospheric temperature not exceeding 90 F and the

barometric pressure not less than 28.25 in. Hg. For power-plant service, the engine rating should be such that it will be capable of delivering at least 10 per cent in excess of its standard commercial rating (sea level or altitude) for a stated period, generally specified for large engines as 2 hr out of any 24 with safe operating

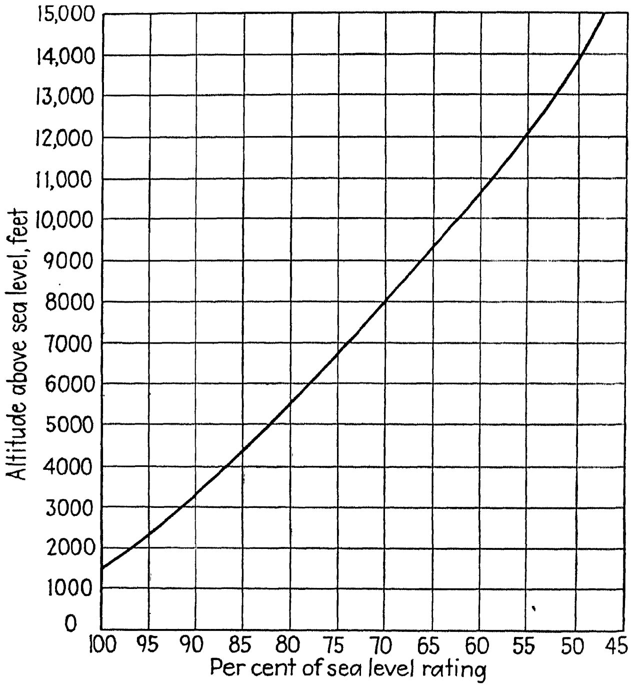


FIG. 64.—Altitude ratings of diesel engines expressed in percentage of sea-level horsepower ratings. (*From D.E.M.A., Standard Practices.*)

temperatures. If such essential auxiliaries as injection air compressor for an air-injection engine, scavenging air pump or blower for two-stroke-cycle engines, or pumps for circulating lubricating oil or piston-cooling oil are separately driven, the horsepower required to drive them should be deducted from the horsepower delivered to the crankshaft coupling in arriving at the net brake horsepower.

The full-load capacity of a diesel engine varies with altitude. The net brake-horsepower rating of a normal nonsupercharged slow- or medium-speed diesel engine at various altitudes, expressed in percentage of the sea-level rating, is shown in Fig. 64.

There are three ratings applicable to engines operating above 750 rpm. These are continuous, intermittent, and peak horsepower capacities. The *continuous-horsepower* rating is the horsepower that the engine is capable of delivering continuously at the stated speed for periods exceeding 24 hr with satisfactory operating and maintenance characteristics. The *intermittent-horsepower* rating is the power that the engine in good operating condition will develop at the stated speed for a period not exceeding 30 min if immediately followed by a load not in excess of the continuous-horsepower rating for at least twice the length of the period of the intermittent load. The *peak-horsepower* rating is the maximum horsepower that the engine in proper adjustment will develop and maintain without drop in speed for at least one minute with a reasonable clean exhaust. The peak-horsepower rating serves only as a guide to indicate the surplus power available in the engine as stipulated by the manufacturer. The foregoing capacities are based upon standard sea-level conditions with a barometric air pressure of 29.92 in. Hg and an air temperature of 68 F. Horsepower corrective figures for altitude, temperature, and humidity applicable to the specified horsepower output capacities are furnished by the manufacturer.

All net horsepower ratings are based on the performance of the engine complete with such auxiliaries as are necessary for the operation of the engine and as specified by the manufacturer for the type of application. These may include air cleaner, fan, pumps, supercharger, scavenging equipment, exhaust silencer, or similar items introducing continuous power losses. Specific installations may, in addition, require optional equipment that may affect the published net horsepower rating of the engine.

68. Supercharging.—The use of supercharging to increase the horsepower developed per cylinder by an engine of a given bore, stroke, and speed is coming into more general use. Essentially, the supercharging process provides more air in the cylinder to support combustion of the fuel by increasing the weight of the initial air charge in the cylinder at the start of compression. With a greater weight of air to support combustion, a larger quan-

tity of fuel can be burned in the engine cylinder during each power stroke without material change in combustion efficiency or mean temperature of the cylinder, piston, or valves. More fuel burned per unit of time means more power produced.

Supercharging is being employed to

1. Increase the rated power output capacity of a given engine size.

2. Produce ratings at high altitudes approximating the unsupercharged sea-level rating.

Theoretically it is possible to increase the horsepower output of a given cylinder considerably by supercharging. From a practical standpoint, however, the theoretical gain in power is not fully realized since a portion of this increase in horsepower is required to drive the compressor which provides the supercharged air. In order to obtain increased horsepower output at the engine shaft through supercharging, it is necessary that the power required for supercharging the engine be less than the increase in shaft horsepower resulting.

The supercharger may be operated by direct drive from the engine crankshaft, by an exhaust-gas turbine such as in the Buchi system, or by an independent power source. Supercharging may also be accomplished by employing the underside of each of the power pistons in a single-acting engine as an air-compressor cylinder.

Practically all two-stroke-cycle engines require compressors or blowers to provide air at sufficient pressure and ample volume to scavenge the cylinders at the end of each working stroke. No supercharging can result from this scavenging air pressure unless provisions are made for creating an air pressure exceeding atmospheric pressure in the cylinder after the exhaust ports have been closed. The scavenging system developed by Sulzer Bros., in which a double row of scavenging ports with valve control of the upper row is employed, permits the cylinder to be charged with air above atmospheric pressure at the time compression starts.

69. Factors Influencing Engine Selection.—Two basic conditions must be satisfied if a suitable engine is to be obtained. These are the specific needs of the purchaser and the most economical means for meeting those needs. The determination of the purchaser's needs involves technical problems dealing with horsepower capacity and number of engines required, fuel and

lubricants, space requirements of engines, and similar matters, while the economic problems involve investments necessary, operating costs, and fixed charges.

The purchaser of an internal-combustion engine wants to buy an economical and dependable unit for producing power. His attention will naturally be directed to those factors which permit him to determine the suitability for his particular requirements

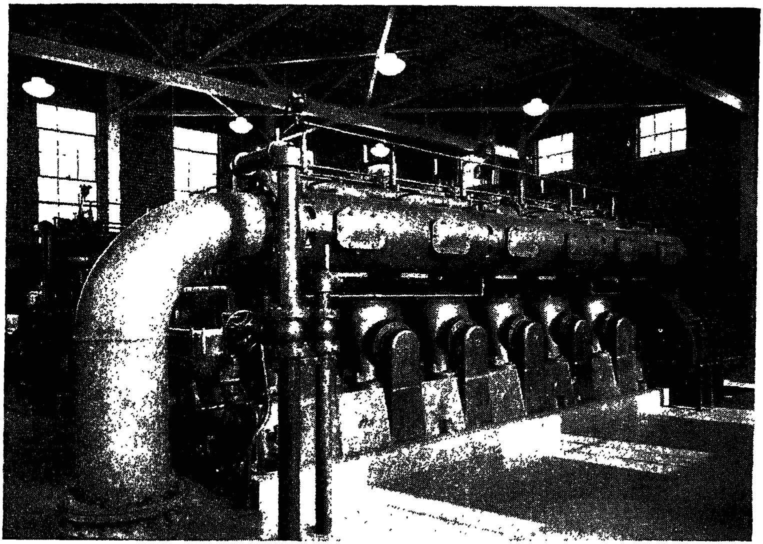


Fig. 65.—Two Fairbanks-Morse 450-hp, two-stroke-cycle, crankcase-scavenging diesel engines installed in a power plant. (*Courtesy of Burns & McDonnell Engineering Company.*)

and the relative dependability and economy of power production for the several makes of engines offered. From the buyer's viewpoint, all elements considered in the selection of an engine are economic, and all design features are studied for the effect they may have on fixed charges and operating and maintenance costs.

Admittedly, the engines considered for any particular application must meet the specific needs of the installation proposed. These requirements will vary, but in general they include horsepower capacity required, space necessary for housing the engine,

and type of fuel and lubricants that can be satisfactorily used in the engine. For example, if a 500-hp unit is needed it is obvious that a 300-hp engine is not large enough for the job and a 1,000-hp unit is entirely too large. Furthermore, if the over-all length of a unit must be limited to 10 ft, a unit 15 ft in length cannot be accommodated.

Conditions do arise where it is impossible to find an engine which exactly meets the requirements of a particular set of conditions. Where such a situation does occur, it becomes necessary to make a compromise and select that unit which most nearly fits the requirements. In reality most engine applications involve compromises in which some of the requirements are not completely met by the engine purchased. This is not unusual, since practically every engineering problem, be it the design of an engine, building, or highway, involves limitations in materials, methods of construction, and cost. These limitations both technical and economic determine the closeness with which the final product approaches the ideal conditions it was supposed to meet.

70. Specifications for the purchase of an internal-combustion engine require much care and thought in their preparation in order that the one proposing to furnish the engine and accessories will know exactly what he must provide. While it may appear to be an easy matter to compile such a specification, those who have had much practice in such work know that important details are easily overlooked. The guiding principle of all specification writing is to set forth plainly what is to be furnished. The more simply these requirements can be presented, the more effective will be the specifications prepared.

In developing specifications, it is necessary to divide the contemplated improvement into three distinct divisions consisting of

1. Equipment and materials furnished and services performed under the specifications being prepared.
2. Equipment and materials furnished and services performed under other specifications.
3. Equipment, materials, and services furnished by the purchaser and not forming a part of any specification prepared.

In the construction of a small diesel power plant one specification may cover the diesel engines, generators, exciters, air filters, mufflers, starting air equipment, interconnecting piping and

their erection. A second specification may provide for the switchboard, while a third may include the cooling tower. The power-plant building, engine foundations, fuel-oil tanks, circulating-water piping, circulating-water pumps, and other accessory equipment and structures would be called for under a fourth specification. In such a condition, it is necessary to be certain that the various specifications are so coordinated that all facilities are provided and that no overlapping of contracts exists.

As a further example, consider that an engine is to be purchased to drive a flour mill. The engine is to be installed in the mill and to drive the mill by means of a belt. It becomes necessary in preparing the specifications to assure that a complete installation will be made. This can be done by providing, in the specification, for the contractor to furnish everything necessary for the proper functioning of the installation through a blanket statement. This procedure may lead to disputes between the contracting parties as to just what is necessary for a complete installation. The specifications can be written in a form setting forth in detail the items of equipment and service to be furnished, or the specification may require the contractor to furnish certain parts of the installation while the purchaser provides some of the remaining parts required for the completed project.

If the latter procedure is followed, it is necessary that care be exercised in setting forth the division of responsibility for supplying equipment, materials, and services in such a manner that there will be no misunderstanding after a contract has been entered into.

After it has been decided just what is to be included in the specifications covering the engine and its accessories, the preparation of the detailed requirements can be undertaken. It is impossible to cover all details which might under various circumstances be required in the specifications. The following list includes a major portion of the items that should be included or specifically omitted in order that the contractor may quote intelligently on the job.

1. Number of engines required.
2. Horsepower per engine or range of engine horsepower ratings considered and whether rating is for continuous or intermittent duty.
3. Type of service for which engine is to be used.

4. Accessories to be included with the engine.
5. Character of fuel to be used.
6. Limitations, if any, as to space available for housing engines.
7. Limitations, if any, on rotative speed, type of design, or cylinder arrangement.
8. Limitations, if any, as to sizes of pieces which can clear available openings.
9. Elevation of plant above sea level.
10. Proximity of railroad siding to plant site.
11. Size and type of crane available to assist in erection.
12. Approximate length of air-intake and exhaust lines required.
13. Type of governor required.
14. Number of electric generators and their individual capacities, mechanical, and electrical characteristics.
15. If unit or units are to drive a-c generators in parallel with existing machines, give name-plate data for existing engines and generators to enable contractor to make necessary parallel operation calculations.
16. Starting equipment required.
17. Fuel- and lubricating-oil tanks and pumps required.
18. Lubricating-oil conditioning equipment needed.
19. Extent of piping to be furnished.
20. Cooling-water facilities either to be provided with the engine or furnished by others.
21. Tests for acceptance—specify where they are to be conducted and whether at the contractor's or the purchaser's expense.
22. Terms and conditions of payment.

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CHAPTER IX

FUEL OIL

Petroleum, either crude or in its various refined forms, is probably the most widely used fuel for power production in the United States. As gasoline, it is used in automobiles, tractors, trucks, airplanes, and high-speed boats. Heavier fractions are burned in compression-ignition engines used in trucks, railroad locomotives, power plants, construction machinery, ships, and some airplanes. The residues serve as boiler fuel in steam power plants, steam locomotives, steam-propelled ships, and for general heating purposes, as well as fuel for large diesel engines in power plants and ships. This chapter will consider these refined petroleum products used for fuel in compression-ignition engines.

71. Fuel Oil.—The American Society of Testing Materials defines fuel oil as,[1]

> any liquid or liquefiable petroleum product burned for the generation of heat in a furnace or firebox, or for the generation of power in an engine, exclusive of oils with a flash point below 100 F (38 C), Tag closed tester, and oils burned in cotton or woolwick burners. Fuel oils in common use fall into one of four classes: (1) *residual fuel oils*, which are topped crude petroleums or viscous residuums obtained in refinery operations; (2) *distillate fuel oils*, which are distillates derived directly or indirectly from crude petroleum; (3) *crude petroleums* and weathered crude petroleums of relatively low commercial value; (4) *blended fuels*, which are mixtures of two or more of the three preceding classes.

Liquid petroleum distillates having viscosities between those of kerosene and lubricating oil are known as *gas oil*.

72. Engine Fuels.—The types and characteristics of fuel oils available for use in internal-combustion engines are largely determined by the method employed for refining the crude petroleum and the blending or mixing of heavy and light fuel-oil fractions. Improvements in refining technique, particularly in the adapta-

[1] Standard Definitions of Terms Relating to Petroleum, A.S.T.M. Designation D 288-39.

tion of cracking during the past 20 years, have progressed to the point where about one-half of the gasoline is being produced in cracking plants. Domestic furnace-oil and diesel-fuel markets, however, were developed as an outlet for the heavier fractions produced in straight-run refining. Cracking-plant operation is designed for high gasoline yield, and the gas oils formerly so abundant in straight-run refining are being converted into gasoline. The introduction of catalytic processing of petroleum hydrocarbons, as typified by the Houdry process, permitted the oil refiner to vary his relative production of gasoline and fuel oils in accordance with the relative market demands for each.[1]

The demand for fuel oil for diesel engines, while increasing rapidly in recent years, represents less than 2 per cent of the total sale of petroleum products in the United States. This is strikingly illustrated by the data covering sale of petroleum products in Table 17.

TABLE 17.—DIESEL-FUEL CONSUMPTION, GALLONS

Year	Diesel fuel	All petroleum products
1934	12,725,000	920,164,000
1935	16,174,000	983,686,000
1936	17,229,000	1,092,754,000
1937	21,904,000	1,169,682,000
1938	21,204,000	1,137,123,000
1939	23,867,000	1,231,076,000

On Jan. 1, 1939, there was slightly over 12 million horsepower of diesel engines installed in the United States. Gasoline engines installed in motor vehicles at that time represented 2,030 million horsepower. Diesel engines thus constituted only 0.6 per cent of the total horsepower of internal-combustion engines, although the fuel used by this group was 1.9 per cent of the total petroleum products sold.

In view of these conditions, it is not surprising to find that a representative of one of the large oil companies in a recent meeting of the Society of Automotive Engineers states:

> The refiner has problems beyond his control which limit his ability to meet, economically, a wide variety of diesel-fuel specifications. If the

[1] MOUNT, W. S., and E. T. SCAFE, The Houdry Process and Its Relation to Diesel Industry, *Proc. O.G.P. Div. A.S.M.E.*, 1940.

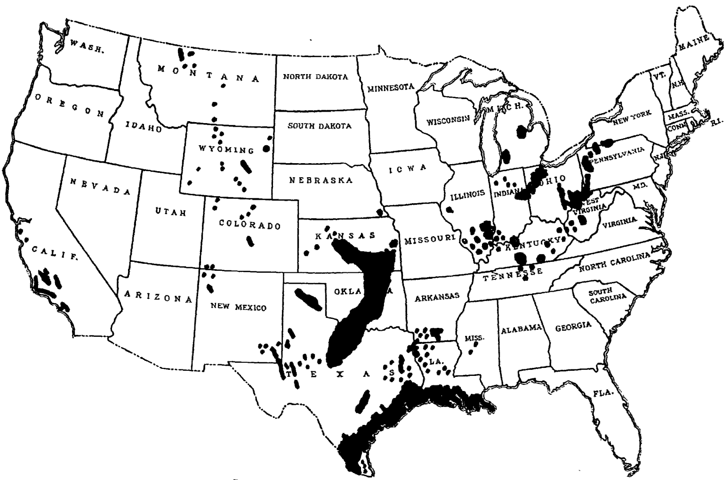


FIG. 66.—Oil fields in the United States.

TABLE 18.—DETAILED REQUIREMENTS FOR FUEL OILS[a] (A.S.T.M. D 396-39T)

Grade of fuel oil[b]	Flash point, deg. F		Pour point, deg. F	Water and sediment, per cent	Carbon residue, per cent	Ash, per cent	Distillation temperatures, deg. F: 10 per cent point	Distillation temperatures, deg. F: 90 per cent point		Distillation temperatures, deg. F: End point	Viscosity, sec.: Saybolt universal (at 100 F)		Viscosity, sec.: Saybolt furol (at 122 F)	
	Min.	Max.	Max.	Max.	Max.	Max.	Max.	Max.	Min.	Max.	Max.	Min.	Max.	Min.
No. 1 {A distillate oil for use in burners requiring a volatile fuel}	100 or legal	165	0	trace	0.05[d] on 10 per cent residuum	....	410	...	...	560[e]				
No. 2 {A distillate oil for use in burners requiring a moderately volatile fuel}	110 or legal	190	10[c]	0.05	0.25[f] on 10 per cent residuum	....	440	600						
No. 3 {A distillate oil for use in burners requiring a low-viscosity fuel}	110 or legal	230	20[c]	0.10	0.15	....	...	675	600[g]	...	45			
No. 5 {An oil for use in burners requiring a medium-viscosity fuel}	130 or legal	...	...	1.00	....	0.10	...	...	...	...	..	50	40	
No. 6 {An oil for use in burners equipped with preheaters permitting a high-viscosity fuel}	150	...	...	2.00[h]	....	....	...	...	...	...	..	..	300	45

[a] Recognizing the necessity for low-sulfur fuel oils used in connection with heat-treatment, nonferrous-metal, glass and ceramic furnaces and other special uses, a sulfur requirement may be specified in accordance with the following table:

Grade of Fuel Oil	Sulfur, max., per cent
No. 1	0.5
No. 2	0.5
No. 3	0.75
No. 5	No limit
No. 6	No limit

Other sulfur limits may be specified only by mutual agreement between the purchaser and the seller.

[b] It is the intent of these classifications that failure to meet any requirement of a given grade does not automatically place an oil in the next lower grade unless in fact it meets all requirements of the lower grade.

[c] Lower or higher pour points may be specified whenever required by conditions of storage or use. However, these specifications shall not require a pour point lower than 0 F under any conditions.

[d] The carbon residue on 10 per cent residuum may be increased to a maximum of 0.12 per cent when this oil is to be used in other than sleeve-type blue-flame burners. This limit may be specified by mutual agreement between the purchaser and the seller.

[e] The maximum end point may be increased to 590 F when this oil is to be used in other than sleeve-type blue-flame burners.

[f] To meet certain burner requirements, the carbon-residue limit on this oil may be reduced to 0.15 per cent on 10 per cent residuum.

[g] This requirement may be waived if the A.P.I. gravity is 26 or lower.

[h] The amount of water by distillation plus the sediment by extraction shall not exceed 2.00 per cent. The amount of sediment by extraction shall not exceed 0.50 per cent. A deduction in quantity shall be made for all water and sediment in excess of 1.0 per cent.

diesel-engine user is to continue to enjoy low-cost fuel, and if the difficulties traceable to the wide variations in the fuels now supplied by the different refiners are to be eliminated, then we must ask for a fuel which will satisfy the refiner as well as the consumer and which will meet the requirements of the diesel engines now in use.

Procurement of fuels for use in diesel engines is an economic problem. Where the engine is capable of burning a wide variety of fuels, the plant operator is able to secure fuel at a relatively low cost year in and year out. On the other hand, the engine requiring a special fuel for satisfactory performance will, by the very nature of the petroleum business, be an engine requiring a relatively costly fuel oil.

73. Fuel Specifications.—Since the development of higher speed diesel engines used in trucks, busses, and other portable service, the oil refiner has been greatly concerned over the many specifications for fuel oil which have followed this development. In an endeavor to secure fuels satisfactory to the refiner, engine designer, and the engine operator, the A.S.T.M. has sponsored the development of a series of standard fuel-oil specifications. The latest available classification of fuel oils prepared by the A.S.T.M. is given in Table 18. Work in connection with this problem is still in progress, and the data contained in Table 18 are tentative and subject to revision.

On the Pacific coast, fuel oil is usually specified and purchased in accordance with Pacific Specifications. These specifications are set forth in Table 19.

In an endeavor to classify the fuel-oil requirements of high-, medium-, and low-speed diesel engines manufactured in the United States, Hubner and Egloff[1] of the Universal Oil Products Company compiled data obtained from 48 manufacturers regarding fuel-oil requirements. Their findings relative to the requirements for fuel oil do not include requirements set up for diesel-electric locomotives and the United States Military Service.

74. Fuel Analysis.—In submitting fuel-oil samples to a testing laboratory for analysis, it is always advisable to specify the tests that should be made. If such a procedure is not followed, the laboratory may make tests that are of little value to the user and may omit other vital information from the laboratory report.

[1] HUBNER, WILLIAM H., and GUSTAV EGLOFF, A Study of Diesel Fuels, *Proc. OGP Division of A.S.M.E.*, 1937, p. 166.

TABLE 19.—PACIFIC SPECIFICATIONS FOR FUEL OILS OF VARIOUS DESIGNATIONS

Grade[a]		Flash point, deg F		Water and sediment,	Viscosity, sec				Distillation temperatures			
					Saybolt universal, at 100 F		Saybolt Furol, at 122 F		10% point		90% point	
Pacific specification number	Description	Max.	Min.	Maxi-	Max.	Min	Max.	Min.	Max.	Min.	Max.	Min.
P.S. 100	A distillate oil for use in stoves, space heaters, burners, and distillate-burning spark-ignition engines requiring volatile fuel, and commonly described stove distillate or stove oil	165	110	0.25					420	350	550	450
P.S. 200	A distillate oil for use in furnaces, burners, diesel or semidiesel engines requiring a low viscosity, moderately volatile fuel, and commonly described as diesel fuel oil, or burner oil		150	0.5	55	35				425		
P.S. 300	A residual oil for use without preheating in furnaces and burners requiring a low-viscosity fuel, and commonly described as light fuel oil, domestic fuel oil, or low-viscosity fuel oil		150	1.0			40					
P.S. 400	A residual oil for use in furnaces and burners equipped with preheaters permitting a high-viscosity fuel, and commonly described as industrial fuel oil, heavy fuel oil, or high-viscosity fuel oil		150	2.0				60				

[a] Intermediate grades of oil to be designated as of the lower Pacific specifications number.

TABLE 20.—MANUFACTURERS' RECOMMENDED FUEL SPECIFICATIONS FOR HIGH-, MEDIUM-, AND LOW-SPEED DIESEL ENGINES

	High-speed (above 1,000 rpm) 25 manufacturers		Medium-speed (500–1,000 rpm) 26 manufacturers		Low-speed (below 500 rpm) 33 manufacturers	
	Spread	Average	Spread	Average	Spread	Average
Viscosity, SU at 100 F:						
Seconds, min	34–40	36	34–75	40	30–75	40
Seconds, max	40–250	78	45–250	87	50–250	106
Gravity, deg A.P.I.:						
Minimum	16–30	26	16–31	25	13–30	24
Maximum	29–45	35	29–45	36	19–45	34
Sulfur, per cent max	0–2.0	0.9	0–2.0	1.0	0–2.0	1.1
Hard asphalt, per cent max	0.05–1.0	0.510	0.05–1.0	0.62	0.3–0.7	0.43
Conradson carbon, per cent max	0–3.0	0.6	0–5.0	1.0	0–10.0	1.7
Ash, per cent max	0.01–0.05	0.03	0–0.1	0.04	0–0.15	0.04
Water and sediment, per cent max	0–2.0	0.4	0–2.0	0.6	0–4.0	0.8
Flash point, deg F min	135–150	148	140–195	152	140–190	151
Pour point, deg F max	....	[b]	....	[b]	....	[b]
Distillation characteristics, deg F:						
10 per cent pt. max	460–475	465	460–510	485		
90 per cent pt. max	660–700	700	660–750	695	660–750	705
E.P. max	650–700	680	....	......	700	700
Ignition quality:[a]						
Cetene No., C.C.R., min	38–45	42	40–50	45	35–40	43
Cetane No., C.C.R., min	45	45	45	45		
Cetane No., delay, min	35–45	37	35–45	40	30–45	37
Diesel index No., min	35–40	38	40–50	45	40	40

[a] Ignition quality values cannot be considered entirely representative because of the differences in nomenclature used for expressing same.

[b] 10 to 15 F below operating temperature.

Many tests have been devised for determining the various properties of fuel oils, but the following are those which will give the information needed for an intelligent study of the fuel.

Gravity, in degrees A.P.I. at 60 F.
Heat content, in Btu per pound, high-heat value.
Flash point, by Pensky-Martin closed tester, in degrees Fahrenheit.
Water and sediment, by means of test centrifuge, in per cent by volume.
Viscosity, in SSU at 100 F, or centistokes at 100 F by kinematic method.
Carbon residue, in per cent by weight by Conradson method.
Ash, in per cent by weight.
Pour point, in degrees Fahrenheit.

Sulfur, in per cent by weight. (In fuels suspected of corrosive sulfur a test for active sulfur should be requested.)

Acidity, reported as *acid*, *neutral*, or *alkaline*.

Ignition quality, in cetane number. (In comparing similar fuels bought from one company, Diesel Index may be a satisfactory substitute.)

Instructions should be given the testing laboratory to make all tests in accordance with methods prescribed by the A.S.T.M.

75. Specific Gravity.—Until recent years, practically all diesel fuel was selected for use largely on the basis of specific gravity. Most plant operators still judge the suitability of a fuel by its specific gravity and are often surprised and puzzled when two different fuels of the same gravity act differently in the same engine. While viscosity and ignition quality influence a fuel's characteristics much more than its specific gravity, nevertheless, the plant operator and designer are still vitally interested in the specific gravity of the fuel oil.

If the specific gravity of the oil is known, the engine builder's guarantees, based upon pounds of oil per brake-horsepower output, can be readily converted into terms of oil purchase which is gallons corrected to 60 F. Likewise, with the specific gravity and temperature of the oil known, the volume of oil purchased can be readily converted to standard volume at 60 F which is generally the basis of volume measurements for sale.

Two specific-gravity scales have been used in the past for petroleum products known as the Baumé scale and the American Petroleum Institute, or A.P.I. scale. Both are expressed in degrees. Under both methods of measurement the specific gravity of a petroleum oil or mixtures of petroleum products is the ratio of the weight of a given volume of the material at 60 F (15.56 C) to the weight of an equal volume of distilled water at the same temperature.

The Baumé scale is in reality two scales, one of which is used for liquids lighter than water and one for liquids heavier than water. For liquids lighter than water,

$$\text{Specific gravity} = \frac{140}{130 + {}^\circ\text{Bé}} \tag{11}$$

For liquids heavier than water,

$$\text{Specific gravity} \quad \frac{145}{145 -}$$

TABLE 21.—OIL-MEASUREMENT DATA AT 60 F[1]

Deg. A.P.I.	Specific gravity	Lb per gal	Approximate heating value		Weight lb per cu ft
			Btu per lb	Btu per gal	
5	1.0366	8.63	18,290	157,840	64.59
6	1.0291	8.57	18,340	157,170	64.12
7	1.0217	8.50	18,390	156,320	63.65
8	1.0143	8.44	18,440	155,340	63.19
9	1.0071	8.39	18,490	155,130	62.78
10	1.0000	8.33	18,540	154,620	62.36
11	0.9930	8.27	18,590	153,740	61.93
12	0.9861	8.22	18,640	153,220	61.50
13	0.9792	8.16	18,690	152,510	61.07
14	0.9725	8.10	18,740	151,790	60.65
15	0.9659	8.05	18,790	151,260	60.24
16	0.9593	7.99	18,840	150,530	59.83
17	0.9529	7.94	18,890	149,980	59.42
18	0.9465	7.89	18,930	149,360	59.03
19	0.9402	7.83	18,980	148,610	58.64
20	0.9340	7.78	19,020	147,980	58.25
21	0.9279	7.73	19,060	147,330	57.87
22	0.9218	7.68	19,110	146,760	57.49
23	0.9159	7.63	19,150	146,110	57.12
24	0.9100	7.58	19,190	145,460	56.75
25	0.9042	7.53	19,230	144,800	56.39
26	0.8984	7.49	19,270	144,330	56.03
27	0.8927	7.44	19,310	143,670	55.68
28	0.8871	7.39	19,350	142,900	55.32
29	0.8816	7.35	19,380	142,440	54.98
30	0.8762	7.30	19,420	141,770	54.64
31	0.8708	7.26	19,450	141,210	54.31
32	0.8654	7.21	19,490	140,520	53.97
33	0.8602	7.17	19,520	139,960	53.64
34	0.8550	7.12	19,560	139,270	53.32
35	0.8498	7.08	19,590	138,690	53.00
36	0.8448	7.04	19,620	138,120	52.68
37	0.8398	7.00	19,650	137,550	52.37
38	0.8348	6.96	19,680	136,970	52.06
39	0.8299	6.92	19,720	136,460	51.76
40	0.8251	6.87	19,750	135,680	51.46
41	0.8203	6.83	19,780	135,090	51.16
42	0.8156	6.79	19,810	134,510	50.86

[1] Courtesy *Power*.

For *liquids lighter than water* the Baumé scale starts at 10° Bé corresponding to a specific gravity of 1.0000, and as the degrees Baumé increase the specific gravity decreases. For *liquids heavier than water* the Baumé scale starts at 0° Bé corresponding to a specific gravity of 1.0000, and as the degrees Baumé increase the specific gravity increases.

The use of two scales for the designation of the specific gravities of petroleum products has led to much confusion. In order to eliminate this constant misunderstanding, the A.P.I. scale was developed. According to the "Smithsonian Physical Tables,"[1]

> In order to avoid confusion and misunderstanding the American Petroleum Institute, the Bureau of Mines, and the Bureau of Standards have agreed that a scale based on the modulus 141.5 shall be used in the United States Petroleum Industry and shall be known as the A.P.I. scale. The United States Baumé scale based on the modulus 140 shall continue to be used for other liquids lighter than water.

Thus the A.P.I. scale given by the formula

$$\text{Specific gravity} = \frac{141.5}{131.5 + \text{°A.P.I.}} \tag{13}$$

serves for petroleum products lighter as well as heavier than water. It is not subject to the limitations of the Baumé scale since inspection of the formula shows that the A.P.I. gravity of water is 10, liquids heavier than water less than 10, and liquids lighter than water greater than 10.

The specific gravity of practically all fuel oil and other petroleum products today is expressed by means of the A.P.I. scale, and standard methods of determining the gravity by test[2] have been developed.

Table 21 is useful for making conversions from the A.P.I. gravity scale to specific gravity, weight per gallon, and weight per cubic foot. Approximate heating values for the different specific-gravity fuels are also given in this table. All data are based upon standard temperature conditions of 60 F. Specific-gravity determinations made at any temperature between 20 and 120 F can be corrected to 60 F by means of Fig. 67, while volume

[1] Eighth revised edition, p. 158.

[2] Standard Method for Test for Gravity of Petroleum and Petroleum Products by Means of the Hydrometer, A.S.T.M. Designation D 287-37.

corrections for the same temperature range can be made by means of Fig. 68. These two figures are based on the National Standard Petroleum Oil Tables, Bur. Standards *Cir.* C 410.

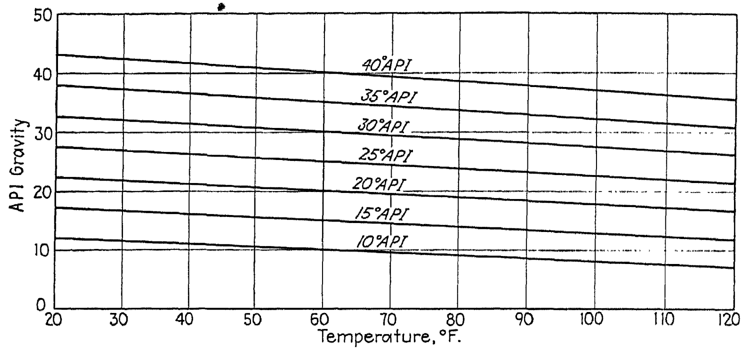


FIG. 67.—Temperature-correction scale for A.P.I. gravity of petroleum products.

76. Heat Content.—The heat content of fuel oil is generally expressed in Btu per pound of fuel on the basis of its high heat value, *i.e.*, the heating value is expressed in terms of the total

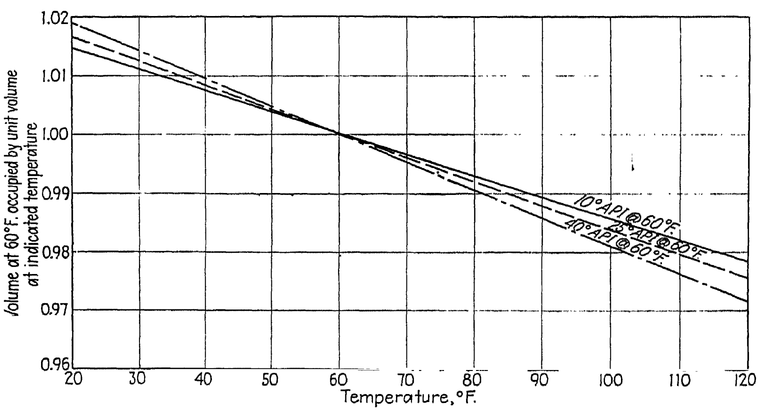


FIG. 68.—Volume-correction scale for petroleum products.

heat generated by the burning of the oil *irrespective of whether this heat is or is not recoverable.* When fuel oil burns, the hydrogen contained in it unites with oxygen to form water. The heat

required to form water vapor cannot be recovered to perform work in the engine cylinder. As a result, while the total heat contained in a pound of fuel oil may be as much as 19,500 Btu, in reality only about 18,500 Btu are available for doing work. It would appear that since all the heat produced by the combustion of fuel oil cannot be recovered it would be advisable to express the heating value of oil on its low heat value. This is not done primarily because it is a simple matter to make determinations of the high heating value, while determinations for the low heating value are more difficult to execute and their accuracy is often questionable. As pointed out in Chap. VII, engine thermal-efficiency computations are based upon the high heat value of liquid fuels.

77. Flash Point.—The flash point of fuel oil is that temperature at which the fuel vapors given off will ignite. Flash point bears no definite relation to the performance of the fuel in the engine. It is primarily needed in testing a fuel in order to determine its relative fire hazard. In this connection, it is always advisable to have tests made by the Pensky-Martin closed tester, since this test procedure gives a better indication of the hazards of fuel oils stored in bulk storage and day fuel tanks.

78. Water and Sediment.—The term *water and sediment,* often called BS&W (bottom sediment and water), is self-explanatory. Sediment should be limited when the fuel is to be used in small high-speed engines. For large slow- and medium-speed units, water and sediment in quantities less than 1 per cent are seldom injurious to the engine. On the other hand, any careful operator should attempt to remove as much foreign matter as possible from the fuel supplied to any engine, either through straining or centrifuging. Excess water is not particularly harmful to an engine, although it may separate out and reach the fuel nozzles in such quantities as to cause misfiring.

79. Viscosity.—All fluids possess a definite resistance to a change of form, which tends to reduce the rate of flow of the liquid. *Viscosity* is the term applied to this internal friction or shearing resistance of fluids.

We are all familiar with the flow characteristics of an asphalt pavement in hot weather when one's heel will make imprints in the surface or automobiles brought to a sudden stop will shove sections of the asphalt from its normal location, resulting in the

formation of waves in the pavement surface. Likewise we are well aware of the slowness with which molasses flows in cold weather. These are all examples of the internal friction within the materials. It is also common experience that heat when applied to materials such as molasses, roofing tar, asphalt, wax, and other similar materials will cause them to "flow like water." When they return to normal temperatures, ranging from 40 to 70 F, most of these substances exhibit only slight tendency to flow.

TABLE 22.—HEAT CONTENT OF PETROLEUM LIQUIDS, BTU PER GALLON[1]

	Degrees A.P.I. at 60 F				
Temp., deg F	10	20	30	40	50
	Specific gravity at 60/60 F				
	1.000	0.9340	0.8762	0.8251	0.7796
0	−105	−102	− 99	− 96	− 93
10	− 73	− 70	− 68	− 66	− 64
20	− 40	− 39	− 37	− 36	− 35
32	0	0	0	0	0
40	+ 27	+ 26	+ 25	+ 24	+ 24
50	61	59	57	55	54
60	95	92	89	86	84
70	130	126	122	118	115
80	165	160	155	150	146
90	201	194	188	182	177
100	237	229	222	215	209
110	273	264	256	248	241
120	310	300	290	281	273
130	347	335	325	315	306
140	384	371	360	349	339
150	422	408	395	383	372
160	460	445	431	418	406
170	499	482	467	453	440
180	538	520	503	488	475
190	577	558	540	524	509

[1] From *Bur. Standards Bull.* 97.

Although a complete understanding of the property of materials known as viscosity involves studies that are not within the scope

of this volume, nevertheless, it is necessary to have a working knowledge of viscosity of fuel and lubricating oils for a proper understanding of problems involved in both the design and operation of the plant.

Viscosity, in its absolute physical conception, is the shear resistance that requires a force of one dyne to move a plate one square centimeter in area separated by the fluid a distance of one centimeter from another surface at a velocity of one centimeter per second. This unit of absolute viscosity is known as a *poise.* Usually absolute viscosity is measured in *centipoises,* where 100 *centipoises* equal one *poise. Kinematic viscosity* is the absolute viscosity divided by the specific gravity and is generally expressed by the ratio u/s in which u = absolute viscosity, poises, and s = specific gravity.

In practical viscosity determinations, it is not possible to measure the absolute viscosity. Instead, viscosity is generally determined by observing the time required for a definite volume of the liquid under definite temperature and head conditions to flow through a tube of small internal diameter.[1] In general, viscosity measurements made in the United States are by means of the Saybolt viscometer, and the viscosity is stated in seconds Saybolt universal (SSU), which is the time in seconds required for a given quantity of the fluid at a given head to run through the viscometer or measuring instrument. *All viscosity determinations must be made at definite temperature conditions, and the temperature at which the viscosity determination was made must be stated. A viscosity determination without knowing the temperature at which it was made is worthless.*

Under certain conditions it is necessary to know the kinematic viscosity of the oil being dealt with. This can be obtained by means of the following equation when the Saybolt universal viscometer reading is known (SSU).

$$\text{Kinematic viscosity} = \frac{u}{s} = 0.0022t - \frac{1.80}{t} \qquad (14)$$

where t = time, seconds, given for viscometer reading.
u/s = kinematic viscosity, stokes.

[1] Viscosity by Means of the Saybolt Viscometer, A.S.T.M. Designation D 88-38.

Equation (14) when used for converting directly from seconds Saybolt universal to centistoke units takes the following form:

$$\frac{u}{s} = 0.22t - \frac{180}{t} \tag{15}$$

where u/s = kinematic viscosity, centistokes.
All that it is necessary to do to obtain the absolute viscosity in centipoises is to multiply both sides of Eq. (15) by s, the specific gravity of the oil under consideration, and we obtain

$$u = 0.22ts - 180\frac{s}{t} \text{ (centipoises)} \tag{16}$$

For purposes of pipe-line calculations it is necessary to convert from metric to English units for the value of absolute viscosity. This conversion is given by the following equation:

$$\mu = 0.000672s\left(0.22t - \frac{180}{t}\right) \tag{17}$$

All letters bear the same values as given in Eqs. (14) to (16). Equation (17) will be of value later in connection with calculations dealing with the loss of head of oil flowing through pipes.

Reference has been made to the fact that any viscosity determination is valueless unless the temperature at which the determination was made is also known. As the temperature of any liquid petroleum product increases, the viscosity either in SSU or centistoke units decreases. The characteristics of this change in viscosity due to temperature are well known, and standard logarithmic charts[1] have been developed for plotting this viscosity-temperature relation. When the temperature and viscosity of a given oil are known for two temperature conditions, a straight line drawn between these two points will show the viscosities at all intermediate temperature conditions.

An example of such a chart is shown in Fig. 69. On this chart are shown the temperature-viscosity relation for two different oils. Oil A has a viscosity of 47.5 SSU at 100 F, while oil B has a viscosity of 550 SSU at the same temperature. Obviously, oil A is much more fluid than B and will offer considerably less

[1] Standard Viscosity-Temperature Charts for Liquid Petroleum Products, A.S.T.M. Designation D 341-39.

friction to flow. On the other hand, water has a viscosity of 31.5 SSU at 60 F, and its viscosity changes but slightly for temperatures between 50 and 120 F.

In addition to knowing the viscosity of the fuel being used in an engine, it is also desirable at times to know what the viscosity of a mixture of two different fuel oils would be. The viscosity-temperature chart can also be used for estimating the viscosity

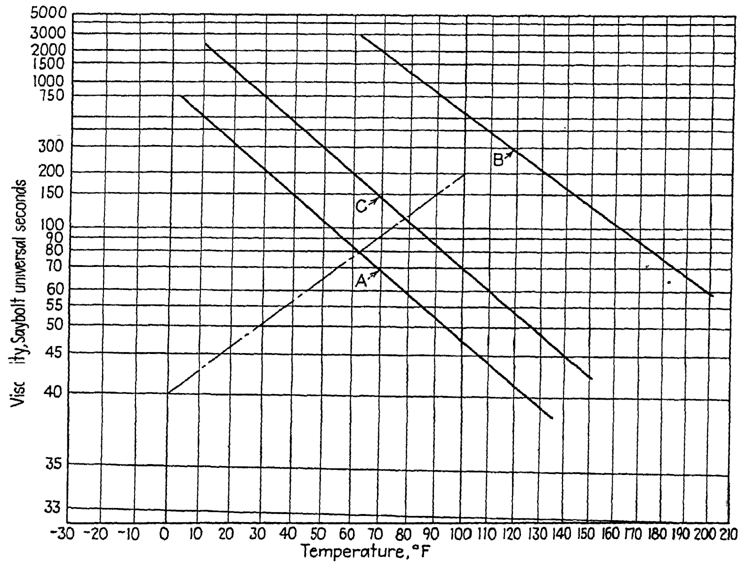


FIG. 69.—Tentative viscosity-temperature chart for liquid petroleum products. (*A.S.T.M.*, D 341-37*T*.)

of mixed oils. If the viscosity of one oil is 40 SSU at 100 F and the viscosity of the second is 200 SSU at the same temperature, viscosities of various proportions of these two oils at 100 F can be determined as follows: On the line for 0 F plot the viscosity of the first oil, namely, 40 SSU; and on the line for 100 F plot the viscosity of the second oil, 200 SSU. Draw a straight line between these two values (see Fig. 69). If these two oils are mixed in equal proportions, the resulting viscosity at 100 F is 64.5 SSU, and if 80 per cent of the mixture is the heavier oil, the resulti viscosity would be 115 SSU.

80. Carbon Residue.—It is important to know the quantity of carbon contained in a fuel oil, and a standard method of test known as the *Conradson carbon-residue* test[1] has been devised for this purpose. While it is somewhat difficult to place a specific interpretation on the results of Conradson tests, it is known that the higher the percentage of carbon in a fuel oil, the poorer the quality of the oil. Small high-speed engines require fuel having a Conradson carbon content of less than ¼ of 1 per cent, while the larger slow- and medium-speed engines are not sensitive to fuels containing much higher percentages of carbon.

81. Ignition quality.[2]—The spontaneous ignition point of a fuel is a function of temperature, pressure, and time. Since it would be difficult to reproduce artificially the temperature, pressure, and time conditions that exist in an engine cylinder, the best apparatus for measuring the ignition quality of a fuel would appear to be an actual diesel cylinder, running under normal operating conditions.

Due to the many uncontrollable variables existing in an actual engine, more reproducible results are obtained if test fuels are compared against standard fuels of known ignition quality run in the same engine and at the same time as the test fuel.

The standard test fuels that have been adopted for measuring the ignition quality of diesel fuels are two hydrocarbon liquids—cetane and alpha-methyl-naphthalene. Cetane has very good ignition qualities while alpha-methyl-naphthalene is very poor in this respect.

The yardstick used for measuring the ignition quality of diesel fuels is the cetane number scale. On this scale, the cetane number of a fuel is the percentage of cetane with alpha-methyl-naphthalene in the standard fuel that matches the ignition quality of the tested fuel. Thus, if a fuel has the same ignition quality as a mixture of 60 per cent cetane in alpha-methyl-naphthalene, the fuel has a cetane number of 60.

Although several methods for measuring ignition quality have been proposed, the "ignition-delay" method using a single-cylinder diesel engine is most widely used. The engine is equipped with a mechanism for varying the compression ratio, other operating conditions being carefully held constant including the delay period. The beginning of the delay period is registered electrically by a contact on the injector plunger. The end of the delay period is also registered electrically by

[1] Standard Method of Test for Carbon Residue of Petroleum Products, Conradson Carbon Residue, A.S.T.M. Designation D 189-39.

[2] Reprinted from "Diesel Operation—Fuel and Lubricants," as published by the Texas Company, New York.

another contact which is actuated by movement of a small diaphragm located in the combustion chamber caused by ignition of the fuel charge.

In determining the cetane number by the ignition-delay method, the engine is first run on the fuel under test and the compression ratio varied until the delay period is adjusted to the standardized length. A fuel of poor ignition quality will require a high compression ratio to shorten the delay period to the standard figure; while a fuel of good ignition quality will have a naturally shorter delay period and compression will have to be lowered to obtain the standardized delay period in the test engine. After the compression ratio required for the test fuel is found and recorded, various mixtures of cetane in alpha-methyl-naphthalene are then tested and on each is determined the compression ratio required to produce the standard delay period. The cetane number of the test fuel is equal to the percentage of cetane in the cetane mixture that required the same compression ratio as the test fuel to produce the standard delay period.

82. Fuel-oil Delivery.—Several methods of fuel delivery are available including delivery by railroad tank car, tank wagon, barge, or pipe line. Barge delivery can be used only at coastal and lake ports or river terminals where the quantities of oil to be used are sufficient to justify shipment by this method. Pipe-line delivery is available when the plant is located adjacent to a pipe-line delivery station or refinery. A major portion of the diesel plants in the United States receive their fuel-oil supply either by tank car or tank wagon.

Standard tank cars vary in capacity from 6,000 to 10,000 gal. Tank wagons vary in size from 600 to 4,500 gal capacity. As pointed out in "Standard Practices,"[1] the minimum capacity of a tank-wagon compartment is approximately 200 gal, and there is usually no reduction in price of tank-wagon delivery for more than one compartment. Some tank-wagon suppliers offer a discount in fuel-oil prices for yearly contracts irrespective of the frequency of delivery. On the other hand, the Petroleum Code does not allow tank-car prices to be quoted if the cars are not unloaded directly into the buyer's storage facilities.

In most instances where diesel engines are installed in electric generating stations and industrial establishments, facilities are

[1] "Standard Practices," Diesel Engine Manufacturers' Association, New York, 1935, p. 55.

provided for storing one or more tank cars of oil, and delivery of oil is taken in carload lots at the plant railroad siding. When the plant is not on a railroad siding, facilities are provided by the purchaser either for pumping direct from the siding to his plant or for truck transportation from the tank car.

83. Fuel-oil Storage Capacity.—Adequate fuel-oil storage is one of the cheapest items that can be purchased for any diesel installation. In view of its cheapness, it is wise to provide too much rather than too little storage capacity.

In determining the amount of storage capacity to provide in any particular case, several factors must be taken into consideration. The factors are the average quantity of oil required per month, method of oil transportation to the plant site, distance from source of oil supply, and fluctuations in fuel prices which are to be encountered from time to time. These several factors will be considered in the order given.

In determining the quantity of oil to be used by the plant, consider that 10 kwhr, or 15 bhp-hr, will be produced per gallon of fuel oil consumed. While practically every diesel engine in power service today will produce more power per gallon than the figures given, nevertheless these values are convenient to remember, easy to apply, and are on the side of safety. Assuming that a plant is generating 10,000 kwhr on the average per month, or 120,000 kwhr annually, we must provide 10,000/10, or 1,000, gal of fuel oil per month. Likewise our annual fuel-oil requirements would be approximately 120,000/10, or 12,000, gal. In other words, this plant would require two carloads of oil annually, the cars being considered as the smallest capacity or 6,000 gal per car.

If this plant is located on a railroad siding where tank cars can be unloaded conveniently, it would be advantageous to buy oil in carload lots. Sufficient storage in this case should be provided to take care of a carload of oil regardless of the amount from a previous carload remaining in stock. In other words, tankage capacity of 12,000 to 15,000 gal would be considered desirable for a plant of the character outlined. On the other hand, if this plant were located in a neighborhood remote from railroad facilities, the storage capacity provided would be influenced by the method of oil delivery, which would undoubtedly be by tank wagon.

The distance that oil must be transported also influences the storage capacity needed, since enough oil must be kept on hand at all times to keep the plant operating even though deliveries are slowed up as a result of strikes, delays in transit, or failure of the supplier to make delivery for any other reason. The diesel-engine plant located adjacent to an oil refinery need have no fears about the availability of fuel. On the other hand, the plant that must ship its fuel 500 to 1,000 miles must always have sufficient oil storage to take care of interruptions and delays in fuel shipment.

One of the larger diesel-electric generating plants supplying utility service is located in a city having a large oil refinery. Everything is favorable for this plant in the matter of oil supply since both oil wells and refinery are within 2 miles of the generating station. It would appear that here was a plant which did not require much bulk oil storage capacity. The facts are that this plant maintains a large bulk oil storage capacity in order to take advantage of fluctuating refinery prices and keep fuel costs low. In this particular case the storage facilities have paid for themselves many times over in the savings made possible through fuel purchases at low prices.

TABLE 23.—CAPACITY OF VERTICAL OIL TANKS, GALLONS

Diameter, ft	Nominal height, ft		
	12	17¾	23½
12	10,080	15,120	20,160
18	12,680	34,020	44,940
24	40,320	60,480	80,220
30	63,000	94,080	125,160
36	90,720	142,800	180,600

84. Fuel-oil Storage Construction.—Bulk storage facilities are arranged for installation either above or below ground. Tanks located aboveground are usually constructed of steel, while buried tanks may be either steel or reinforced concrete. The type and location of fuel-storage tanks will be influenced both by the arrangement of plant equipment and the regulations of the National Board of Fire Underwriters.[1]

[1] "Containers for Storing and Handling Flammable Liquids," National Board of Fire Underwriters Pamphlet No. 30, edition of 1939.

The A.P.I. has issued standards covering vertical storage tanks, *i.e.*, tanks with vertical axes.[1] A number of standard sizes of tanks have been established by the A.P.I., although those in the

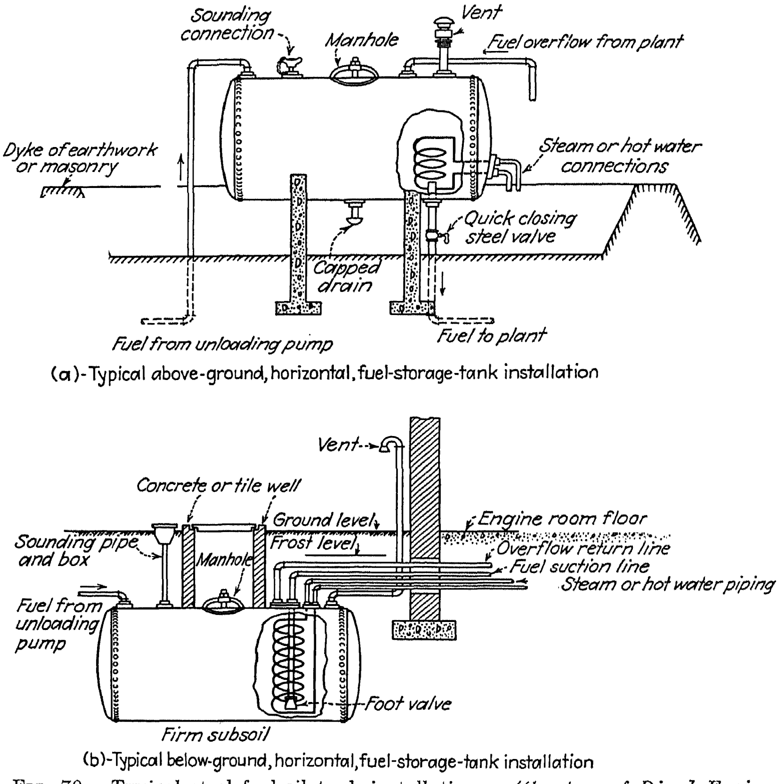


FIG. 70.—Typical steel fuel-oil tank installations. (*Courtesy of Diesel Engine Manufacturing Association and Diesel Publications.*)

table on page 153 are probably the sizes that will be found more often in diesel installations.

While there is no standard for the construction of horizontal fuel tanks, the sizes given in Table 24 are considered standard with most tank builders.

[1] "A.P.I. Specifications for Standard Tanks with Riveted Shells," A.P.I. Standard No. 12-A, 4th ed., January, 1934.

TABLE 24.—CAPACITY OF HORIZONTAL OIL TANKS, GALLONS

Length of tank, ft	Tank diameter, ft	
	8	10
17	......	10,000
22	......	15,000
27	10,000	20,000
40	15,000	

Typical tank assemblies showing steel tanks installed either above or below ground are contained in Fig. 70. The construction of a typical reinforced-concrete tank of 25,000 gal capacity for installation below ground is shown in Fig. 71. In designing reinforced-concrete tanks, care should be taken to see that no oil-soluble materials are used in expansion joints or leakage of the tank will occur. It is advisable to coat the inside of a concrete tank with an oil-proofing compound such as Ferrotex or a water-glass solution.

85. Cleaning Fuel Oil.—Fuel oil before being used in a diesel engine should be cleaned of impurities such as sediment, foreign matter that may clog fuel-injection pumps and nozzles, water, and other materials that may be injurious either to the injection system or cylinder. As a safeguard in this respect, engine manufacturers provide each engine with a strainer type of fuel-oil filter which is installed in the fuel-oil line ahead of the engine fuel pump.

In addition to the cleaning accomplished by this strainer furnished with the engine, it is often found necessary to remove water, sludge, and other foreign matter from the fuel oil prior to its coming in contact with the engine filter. This oil conditioning may be accomplished in several ways depending upon the character of the fuel oil used and the type of impurities that must be removed. Impurities in the fuel oil will not readily settle out when the viscosity of the oil at normal storage temperatures exceeds 150 SSU.

The most common means for cleaning fuel oil is centrifuging. If fuel oil contains over 0.08 per cent of impurities, it is considered advisable to purify it. Centrifuge equipment will usually reduce this impurities content to about 0.04 per cent and will eliminate

practically all the water present if the fuel oil is centrifuged at a low temperature. It is common practice to heat fuel oil before centrifuging; in some instances the temperature of the oil is raised to as much as 150 F before passing through the centrifuge.

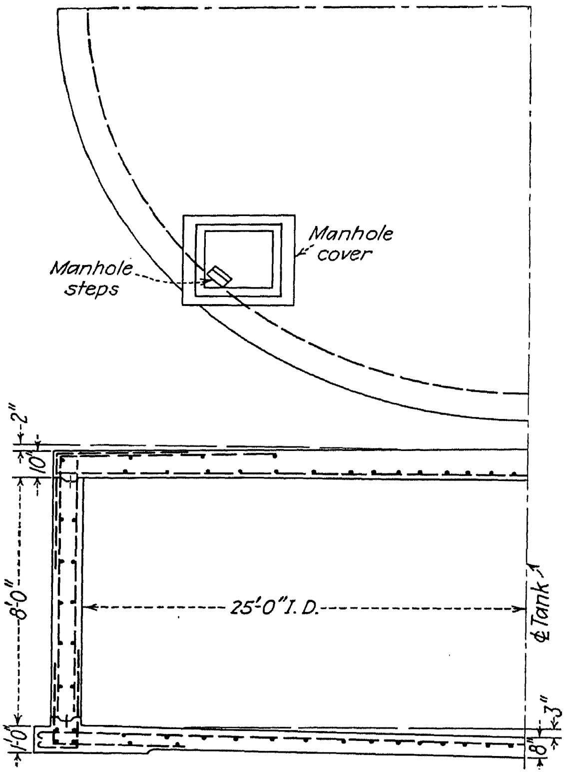


FIG. 71.—25,000-gal reinforced-concrete underground fuel-storage tank. (*Courtesy of Burns & McDonnell Engineering Company.*)

Recent experiences indicate that fuel oils have often been heated to an excessive temperature prior to the cleaning by centrifuging equipment. It has been found in many instances that when oils are heated to a temperature of 150 F before centrifuging the water present goes into solution in the oil and is not removed

by the cleaning process. Furthermore, it has been found that in one instance where heavy residual oils were being cleaned a much better cleaning operation was performed by the centrifuge if the oil was heated to only 110 F instead of the former practice of heating to 150 F before centrifuging. Where crude oils are being centrifuged at 70 F, it has been found that a better job of cleaning

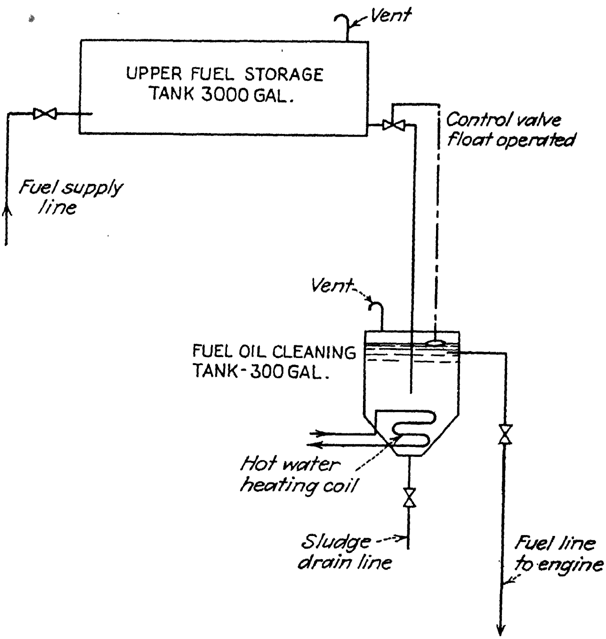


FIG. 72.—Fuel-oil conditioning system employing heating and settling.

is being done than was possible when the cleaning operation was carried out at higher temperatures.

As the centrifuging temperature is reduced, the capacity of the centrifuge is likewise reduced. While this might appear to be a disadvantage, the ability to secure clean fuel at a reduced rate is far superior to the inability to secure clean fuel oil at a more rapid centrifuging rate. Additional capacity in a centrifuge is a relatively inexpensive item, and money invested in a larger centrifuge is a wise expenditure.

Another method of fuel cleaning utilizing heat and settling has been used successfully in one large plant installation. In this particular instance, a fuel oil of approximately 11 A.P.I. gravity and having a viscosity of approximately 450 SSU at 100 F is used. The arrangement of settling and heating facilities employed in this installation is shown in Fig. 72. A day storage tank of 3,000 gal is mounted above the settling tank of 300 gal capacity which is constructed with a hopper bottom and equipped with a 2-in. OD copper heating coil. A float-operated valve is installed in the oil line connecting the two tanks and maintains a constant oil level in the lower or settling tank. The fuel-supply line to the engine is taken off near the top of the lower tank as shown. The temperature of the oil in the settling tank is maintained at approximately 130 F by means of hot water obtained by diverting a portion of the cooling water after it leaves the engine through a heating coil in the engine muffler and thence through the copper coil in the settling tank. Provisions are made for periodical draining of the water and sludge which accumulates in the bottom of the settling tank.

Exhaustive tests in this particular instance indicate that cylinder-liner wear is no greater with this type of fuel cleaning than it was when the fuel oil was centrifuged.

86. Flow of Oil in Pipes.—In the operation of a diesel power plant, fuel oil must be transferred from tank car or tank wagon to the bulk storage facilities at the plant, from the bulk storage tanks to the day tanks for the various engines, and from the day tanks to the engines as their needs require. This transportation problem requires that suitable pipe, valves, fittings, pumps, and other accessories be provided for its accomplishment. The piping system must not only operate, it must operate economically, *i.e.*, the fixed annual charges on the investment in the piping system together with the cost of power required to pump the oil through the system must be a minimum.

The full realization of the economies possible in the operation of the fuel-handling and distribution system requires that the person planning the fuel-handling facilities possess a working knowledge of the hydraulic calculations necessary. Several methods have been evolved for determining friction losses in pipes carrying petroleum products, and while each method of calculation possesses certain advantages, the following method is

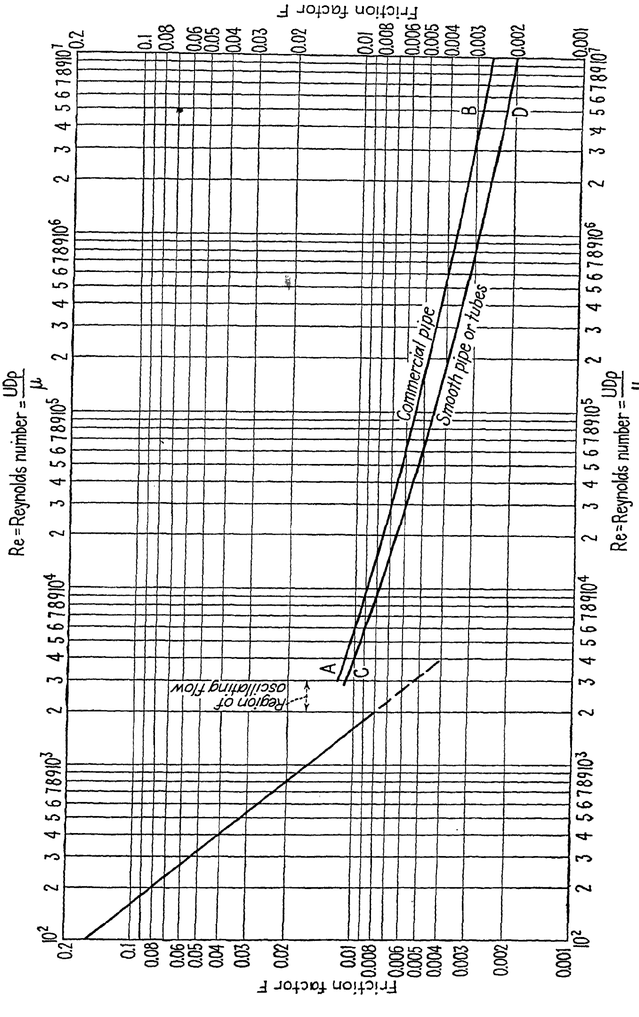


FIG. 73.—Chart showing relationship between Reynolds number and friction factor in oil pipe lines. (*Courtesy of Oil and Gas Journal.*)

preferred by the author. It involves a series of calculations which are necessary to determine (1) the friction factor for the oil and the pipe line under consideration, and (2) the actual pressure drop in the pipe line.

In making these calculations it is necessary to know whether the flow of the oil is *streamline,* in which each particle of oil moves parallel to the axis of the pipe, or whether the flow is *turbulent,* in which particles of oil move in many directions as they progress through the pipe. In changing from streamline to turbulent flow, there occurs a region of oscillating flow where it is possible for "water hammer" to occur in the pipe line with the oil flowing through it at constant velocity. This is known as the region of *critical velocity.* The regions of streamline, oscillating, and turbulent flow are shown on Fig. 73 and are dependent upon the *Reynolds number* for the flow conditions being considered.

The straight portion of the curve, Fig. 73, for Reynolds numbers between 100 and 2,000 represents the region of streamline flow and applies equally to all types of pipes and tubes. The region of oscillating flow occurs for Reynolds numbers between 2,000 and 3,000, and it is this region which should be avoided in determining the relationship of pipe diameter, velocity of flow, density, and viscosity of the oil; see Eq. (18). When Reynolds numbers exceed 3,000, the oil flow becomes turbulent. The upper curve *AB* represents friction factors for commercial pipe, while the curve *CD* applies to smooth pipes and tubes.

For purposes of illustration, let us consider that an oil having an A.P.I. gravity of 25 and a viscosity of 200 SSU at 60 F is to be pumped through a 2-in. nominal diameter line at a rate of 10 gpm. It is desired to find the pressure drop per 100 ft of this line when standard-weight steel pipe is being used.

From Table 21, it is seen that an oil of 25 A.P.I. gravity has

$$\text{Specific gravity} = 0.9042$$
$$\text{Weight per cubic foot} = 56.39 \text{ lb} = \rho.$$

From Table 25, the velocity of 10 gpm flowing through a 2-in. standard-weight iron pipe is found to be 0.96 fps.

The viscosity of the oil in English units is determined by substituting known values in Eq. (17) as follows:

$$\mu = 0.000672 \times \text{sp. gr.} \left(0.22t - \right.$$

TABLE 25.—VELOCITY OF FLOW THROUGH STANDARD WEIGHT IRON PIPE, FPS

Rate of flow, gpm	Nominal pipe diameter, in.					
	1	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	4
	Actual ID					
	1.049	1.610	2.067	2.469	3.068	4.026
10	3.71	1.58	0.96	0.67	0.43	0.25
20	7.42	3.15	1.91	1.34	0.87	0.50
30	11.14	4.73	2.87	2.01	1.30	0.76
40	14.85	6.30	3.82	2.68	1.74	1.01
50	18.56	7.88	4.78	3.35	2.17	1.26
60	22.28	9.45	5.74	4.02	2.60	1.51
70	25.99	11.02	6.69	4.69	3.04	1.77
80	29.70	12.59	7.65	5.36	3.47	2.02
90	33.41	14.17	8.60	6.03	3.90	2.27

NOTE.—The values given in this table can be used to find the velocity of flow at any flow rate. For example, while the velocity of 40 gpm flowing in a 2-in. nominal ID pipe is 3.82 fps, the velocity for 4 gpm in the same pipe would be 0.382 fps. For a flow of 44 gpm in the same pipe, the velocity would be 3.82 + 0.382 = 4.202 fps.

Where μ = viscosity, pounds per square foot per second.
t = SSU value.

Substituting

$$\mu = 0.000672 \times 0.9042(0.22 \times 200 - {}^{180}/_{200})$$
$$= 0.0262$$

It next becomes necessary to determine the Reynolds number for the oil under consideration. This factor can be calculated by means of the equation

$$DU\rho \qquad (18)$$

where D = diameter of pipe, feet = 0.1723.
U = velocity, feet per second = 0.96.
ρ = weight per cubic foot = 56.39.
μ = viscosity = 0.0262.

Substituting in Eq. (18) we obtain

$$R_e = \frac{0.1723 \times 0.96 \times 56.39}{0.0262} = 356$$

From Fig. 73 for a Reynolds number of 356 the friction factor F is 0.045.

The pressure drop in the line can now be calculated by means of the equation

$$P = \frac{2FLU^2\rho}{144gD} \tag{19}$$

where F = 0.045 = friction factor from Fig. 73.
L = 100 ft = length of pipe line.
g = 32.2.
P = pressure drop, psi.

And the other symbols in the equation have values as previously given.

Substituting in Eq. (19), we obtain

$$P = \frac{2 \times 0.045 \times 100 \times (0.96)^2 \times 56.39}{144 \times 32.2 \times 0.1723} = 0.585 \text{ psi}$$

Thus the pressure drop per 100 ft of pipe line is 0.585 psi. The pressure drop for any length of line of this diameter operating under the conditions set forth can be readily determined by direct proportion.

In the problem just given the oil was considered to have a viscosity of 200 SSU at 60 F. Let us consider what would happen if this oil were heated to 100 F before pumping. From the curve (C) plotted on the Viscosity-temperature Chart, Fig. 69, for this particular oil, it is seen that when the temperature increases to 100 F the viscosity decreases to 70 SSU. Since the temperature has changed, the density of the oil has also changed, and it becomes necessary to find the specific gravity and weight per cubic foot of the oil at the higher temperature. From Fig. 68 it is seen that the volume of 25 A.P.I. gravity oil at 100 F is 0.9838 of its volume at 60 F. Using this value for the volume and correcting, the new specific gravity and weight per cubic foot become

$$\text{Specific gravity} = 0.9838 \times 0.9042 = 0.889$$
$$\rho = 0.9838 \times 56.39 = 55.40 \text{ lb per cu ft}$$

And from Eq. (17)

$$\mu = 0.000672 \times 0.889 \times (0.22 \times 70 - {}^{180}\!/\!_{70}) = 0.00766$$

while

$$R_e = \frac{0.1723 \times 0.96 \times 55.40}{0.00766} = 1.195$$

From Fig. 73 it is seen that this is practically at the point where it can change from streamline to oscillating flow.

$$F = 0.008$$

and

$$P = \frac{2 \times 0.008 \times 100 \times 0.96^2 \times 55.40}{144 \times 32.2 \times 0.1723} = 0.15 \text{ psi}$$

If the quantity of oil to be pumped is increased to 50 gpm with the viscosity remaining at 70 SSU and the temperature at 100 F, the only value that has changed is the oil velocity. This increased velocity increases the Reynolds number since

$$U = 4.78 \text{ fps for 50-gpm flow}$$

$$R_e = \frac{0.1723 \times 4.78 \times 55.40}{0.00766} = 5.$$

From Fig. 73 it is seen that the flow is now turbulent and for rough pipe

$$F = 0.01$$

and

The foregoing three conditions of flow have been calculated for the same oil at different temperatures and different rates of flow in order to show the changes necessary in the computations to take into account these variations.

So far no consideration has been given to pipe entrance and exit losses nor has any consideration been given to losses through valves and fittings. While these losses can generally be neglected for lines over 5 miles in length, the length of oil lines under consideration for diesel plants make it necessary to allow for losses through valves, fittings, entrances, and exits. There is no

rational basis for estimating losses of this type, and at best only approximate values can be obtained.

For estimating purposes the loss through fittings can be taken as twice the equivalent length of straight pipe as determined

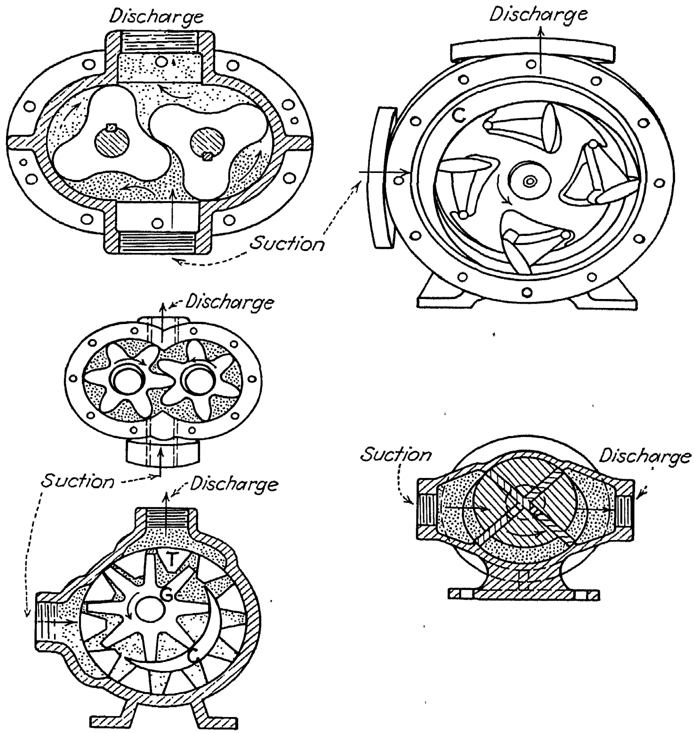


FIG. 74.—Typical construction details of rotary-type pumps. (*Courtesy of Power.*)

from Fig. 101, Chap. XII. This chart shows the equivalent length of straight pipe to equal the friction loss through a 2-in. 90-deg screwed elbow is 5.5 ft, and for estimating the loss with oil flowing the equivalent length of pipe would be 2×5.5, or 11, ft. Friction losses for other types of fittings and valves can be estimated in a like manner.

87. Fuel-oil Pumps.—Pumps used in handling oil are practically all of the rotary type. Rotary pumps are positive-displacement units suitable for handling thick and viscous liquids including petroleum products. They are self-priming owing to their positive-displacement action and have been developed to cover all pressures up to 1,000 psi, capacities as great as 35,000 gpm, viscosities as high as 500,000 SSU, and temperatures up to 850 F.

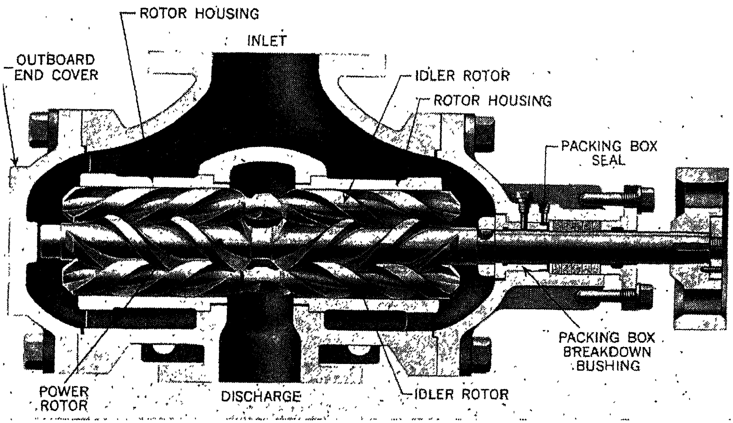


FIG. 75.—Sectional view through gear-type oil pump. (*Courtesy of De Laval Steam Turbine Company.*)

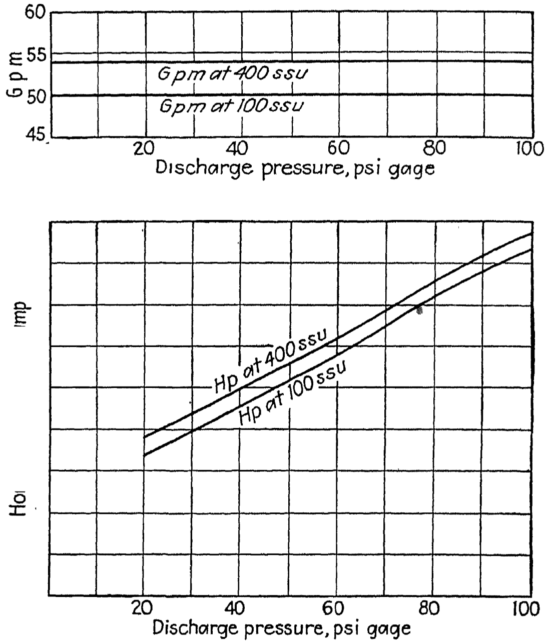
Rotary pumps are designed in numerous ways to effect positive displacement; Fig. 74 shows some of the types manufactured.

As pointed out in the Standards of the Hydraulic Institute,

Positive-displacement rotary pumps in general are constructed with a minimum of clearance between the various parts commensurate with manufacturing methods and various operating conditions. The clearance differs in rotary pumps owing to individual characteristics of the different types of design and naturally the volumetric efficiency of the pump is affected by these clearances.

When operating with little or no load on suction or discharge, pumps of this type will deliver approximately 100 per cent of their displacement. When operating under other conditions the delivery will be less than the pump displacement being affected by speed, vacuum at suction, discharge pressure, viscosity of material, and air or entrained gases in material being pumped.

The effects of viscosity and discharge pressure on the capacity of a rotary pump are shown in Fig. 76 for a unit rated at 50 gpm at 100 SSU. This figure also shows the horsepower required to drive the pump at various discharge pressures. It will be noted that, in this instance, the manufacturer shows no variation



G. 76.—Characteristics of gear-type pumps.

in quantity of oil of a given viscosity pumped when the discharge pressure varied from 0 to 100 psi.

88. Fuel Metering.—In order to keep account of the oil used in the operation of a diesel power plant it is necessary that suitable meters be installed. Meters are usually provided in the oil-flow line between the bulk and day storage tanks, and in some instances between the day tanks and each engine. Meters placed in the latter location may give readings that are rather inaccurate because of the relatively slow rate of flow between the day tank

and the engine. When the day tank is located below the engine with a pump forcing the oil from the tank to the engine, a meter placed in the line between the pump and the engine will not meter the true fuel consumption of the engine. This comes about because the pump, operating at a constant pumping rate, is constantly delivering to the engine more oil than is required by it. The excess oil flows back to the day tank through an overflow line from the engine fuel-pump reservoir to the day tank.

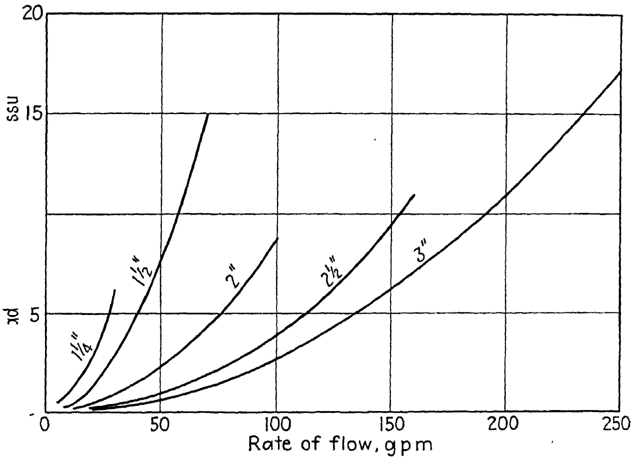


FIG. 77.—Head loss through oil meters. For other viscosities, head loss = where u/s kinematic viscosity in centistokes. (*Courtesy of*
80
Worthington Pump and Machinery Corporation.)

When the day tank is located above the engine and the oil flows by gravity to the fuel pumps on the engine, the rate of flow from the tank to the engine is determined by the load on the engine. For example a 1,000-hp engine at full load, burning a 24 A.P.I. gravity oil, will only use 0.88 gpm, while at one-half load it will only consume 0.47 gpm. It is difficult to get meters that will have any degree of accuracy for such low flows.

Since meters used for measuring fuel oil operate by measuring the volume of oil passing through the meter, and since the volume of oil varies considerably with temperature, it is necessary to make adjustments for this volume variation in meter readings to

obtain an accurate determination of fuel consumption. Some fuel meters are provided with means for determining the temperature of oil flowing through the meter, as well as means for adjusting readings to compensate for this temperature variation. Even with meters equipped to compensate for temperature variations, it is always advisable to check fuel-oil consumption periodically against the volume removed from the bulk storage tanks.

Head loss through one make of fuel-oil meter for several meter sizes is given in Fig. 77. Different makes of meters show considerable variation in the loss through them, and it is always advisable to check the head loss for the particular meter used.

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CHAPTER X

GAS FUEL

Hydrocarbons, compounds of hydrogen and carbon, form the major heat-producing portions of gas and oil fuels. There are many hydrocarbons divided into several series with definite relationships among the members of each series. Only four hydrocarbons, all of the methane series, are normally found in gas fuels. These are

Name	Chemical symbol	Boiling temperature at atmospheric pressure	
		Deg C	Deg F
Methane	CH_4	-164.7	−264.5
Ethane..	C_2H_6	- 86	−122.8
Propane.	C_3H_8	- 39	− 38.2
Butane..	C_4H_{10}	1	33.8

The boiling temperature of these hydrocarbons is influenced by the number of carbon atoms per molecule; as the number of carbon atoms increases, the boiling point is raised.

Gas fuels used in internal-combustion engines may be natural gas, propane, butane, sewage gas, or any of several manufactured or by-product gases. Of these, natural gas, propane, butane, and blast-furnace gas are the most important commercially, although sewage gas is being widely used as a power source in the operation of sewage-disposal plants.

89. Natural Gas.—Natural gas is a mixture of hydrocarbons in which the predominating constituent is usually methane (CH_4) with lesser amounts of ethane (C_2H_6), carbon dioxide (CO_2), and nitrogen (N_2). In most natural gas, methane constitutes over 70 per cent of the volume, and in some instances it may exceed 99 per cent of the gas volume. Ethane usually constitutes 10 to

20 per cent of the volume. Some natural-gas wells have produced essentially carbon dioxide (over 97 per cent in one instance), while others have been largely nitrogen. The presence of large quantities of carbon dioxide or nitrogen in the gas makes it unsuited for fuel. Consequently these natural gases are used only for chemical processes.

Analyses of typical natural-gas samples from widely separated fields as well as analyses of gas supplies in four cities are given in Table 26.

TABLE 26.—TYPICAL NATURAL-GAS ANALYSES

No.	Source	Per cent by volume				Specific gravity	Btu per cu ft at 60 F and 30 in. Hg	
		Methane (CH_4)	Ethane (C_2H_6)	Carbon dioxide (CO_2)	Nitrogen (N_2)		Gross	Net
1	Amarillo, Tex.	72.94	18.96	0.39	7.71	0.68	1086	978
2	Monroe, La.	94.7	2.8	0.2	2.3	0.58	1019	913
3	Ashland, Ky.	75.0	24.0	0.0	1.0	0.68	1197	1079
4	Hugoton, Kan.	68.86	17.51	0.10	13.33	0.70	1018	917
5	Columbus, Ohio	80.4	18.1	0.0	1.5	0.65	1147	1031
6	Birmingham, Ala.	90.0	5.0	0.0	5.0	0.60	1002	904
7	Pittsburgh, Pa.	83.4	15.8	0.0	0.8	0.61	1129	1021
8	Los Angeles, Calif.	77.5	16.0	6.5	0.0	0.70	1073	971
9	Kansas City, Mo.	84.1	6.7	0.8	8.4	0.63	974	879

NOTE.—Specific gravity of gas referred to air.

Numbers 1 to 5 are analyses for typical natural gas from fields as reported in "Economic Utilization of Natural Gas," by Davis, Ihrig, Sabin, and Terry, *Trans. A.S.M.E.*, 1931, p. 388.

Numbers 6 to 9 are the average composition of gas served consumers as reported in "Fuel-flue Gases," p. 20, published by American Gas Association.

The principal gas fields in the United States are the Appalachian, comprising parts of Tennessee, Kentucky, West Virginia, Ohio, Pennsylvania, and New York; the Mid-Continent area in Kansas and Oklahoma; and in the states of Indiana, Illinois, Arkansas, Louisiana, Texas, Wyoming, and California; but gas is also found in many other states as shown by the map in Fig. 78. This wide distribution of natural gas throughout a considerable portion of the country, together with construction of gas pipe lines from the major producing fields to most of the large metropolitan centers, has resulted in widespread use of natural gas for both residential and industrial service.

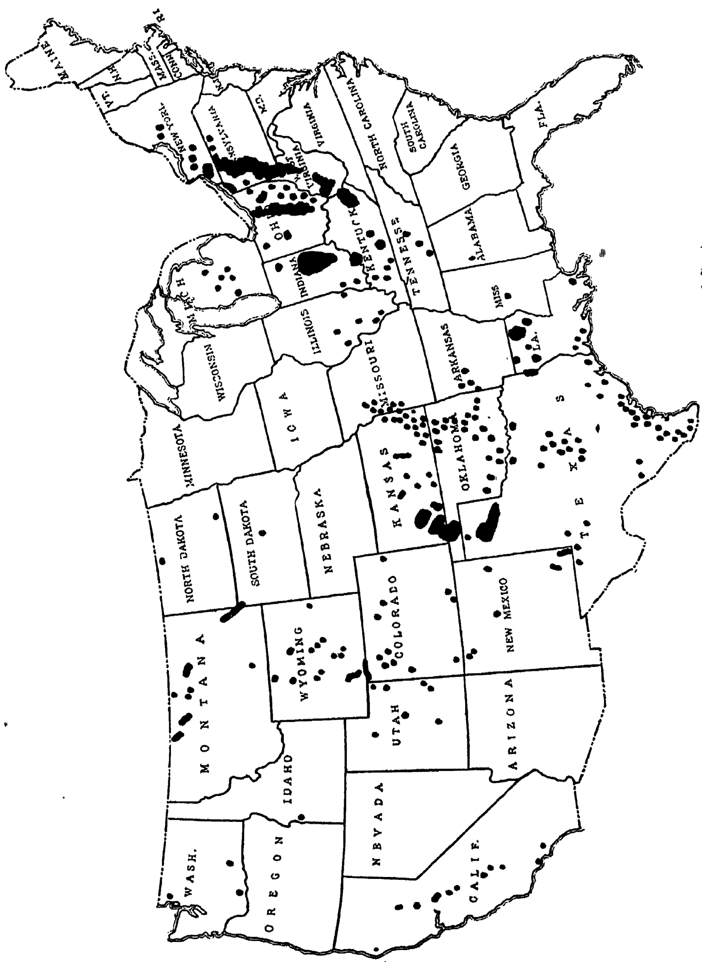

FIG. 78.—Principal natural gas fields in the United States.

90. Gas Temperatures in Pipe Lines.—Occasionally it is desirable to know the temperature at which gas is flowing in a pipe line. At Kansas City, Mo., the average gas temperature is 58 F and ranges from 45 to 85 F. In northern Louisiana the temperatures will range from 65 to 100 F at the point of delivery. Gas temperatures will be influenced by the ground temperature at the point below the ground surface where the line is laid probably somewhat in the same manner as the temperature of oil being pumped through a pipe line varies with ground temperature.

91. Propane and Butane.—Propane and butane are marketed today in liquid form under pressure for both heating and power purposes. The original commercial source for these gases was the natural gasoline plant designed to extract gasoline from natural gas. In the process of removing the gasoline, propane and butane were also removed from the gas. The disposal of these was a considerable problem in the earlier days of natural gasoline production, and they were usually burned as waste products. When a demand arose in the United States for liquefied petroleum gas, these two gases fitted the requirements admirably.

Demand for both propane and butane in liquid form became so great that the natural gasoline became the by-product in the recovery plant. Propane and butane are also extracted in quantity from gases produced in petroleum-refinery operations instead of being burned with other refuse fuels in the refining process.

Typical analyses of commercial propane and butane are contained in Table 27.

TABLE 27.—TYPICAL ANALYSES OF PROPANE AND BUTANE[a]

Name	Constituents of gas, per cent by volume					Specific gravity air = 1.00	Btu per cu ft at 60 F and 30 in. Hg	
	C_2H_6	C_3H_6	C_3H_8	C_4H_8	C_4H_{10}		Gross	Net
Propane from natural gas	2.2	....	97.3	....	0.5	1.55	2558	2358
Propane from refinery gas	2.0	24.3	72.9	....	0.8	1.77	2504	2316
Butane from natural gas	...	....	6.0	....	94.0[b]	2.04	3210	2961
Butane from refinery gas	...	....	5.0	28.3	66.7[c]	2.00	3184	2935

[a] Table prepared from "Fuel-flue Gases" by American Gas Association.
[b] 23.3 per cent isobutane, 70.7 per cent *n*-butane.
[c] 16.5 per cent isobutane, 50.2 per cent *n*-butane.

The physical properties of commercial propane and butane shown in Table 28 are average values.

TABLE 28.—PHYSICAL PROPERTIES OF COMMERCIAL PROPANE AND BUTANE

Physical properties	Propane	Butane
Vapor pressure, psi gauge:		
At 70 F	124	31
At 100 F	192	59
At 130 F	286	97
Specific gravity of liquid (60/60 F)	0.509	0.582
Initial boiling point at 14.7 psi abs., degree Fahrenheit	−51	15
Weight per gallon liquid at 60 F, pounds	4.24	4.84
Cubic feet of gas at 60 F, 30 in. Hg per gal of liquid at 60 F	36.28	31.46
Specific heat of gas, Btu per lb per deg F at 60 F (cp)	0.404	0.382
Latent heat of vaporization at boiling point:		
Btu per pound	185	167
Btu per gallon	785	808

92. Sewage Gas.—Sewage gas, a product resulting from the decomposition of human, animal, and industrial waste, has become of some importance as a power source for the operation of sewage-disposal plants. The major portion of sewage gas consists of methane (CH_4), ranging from 60 to as high as 80 per cent, and carbon dioxide (CO_2), varying from 14 to 35 per cent. Other gases, such as hydrogen, oxygen, nitrogen, carbon monoxide (CO), or hydrogen sulfide (H_2S), may be present in varying amounts. Considerable variation exists in the composition of sewage gas produced by various plants as well as by any single plant. This variation in composition is plainly apparent from the analyses contained in Table 29. In general, sewage gas can be considered to consist of two-thirds methane (CH_4) and one-third carbon dioxide (CO_2).

Production of sewage gas is extremely variable and is dependent upon the temperature at which sewage sludge is decomposed, the density of the sludge, and the character of the sludge. Gas production may vary from over 300 per cent of the average daily production to less than 10 per cent of the average where sludge-digestion tanks are not heated. Where sludge-digestion tanks are heated, however, the maximum rate of gas production may be only approximately three times the minimum production.

TABLE 29.—COMPOSITION OF TYPICAL SEWAGE GASES

Municipality	Per cent by volume[a]						Btu per cu ft heating value at 60 F and 30 in. Hg	
	CH_4	H_2	CO	CO_2	O_2	N_2	Gross	Net
Imhoff tanks:								
Chicago, Ill., Calumet Plant	76.6			7	0.5	8.2	776	699
Dayton, Ohio	76.6			0	0.4	9.0	776	699
Decatur, Ill.	69.0			8	0.1	14.1	699	630
Stuttgart, Germany	75.5	4.7		0		4.7	780	702
Separate sludge-digestion tanks:								
Antigo, Wis.	62.0	2.6		4	0.6	3.4	636	573
Aurora, Ill.	51.8		2.1	3	0.4	13.4	532	480
Baltimore, Md.	70.5			5	0.2	2.8	714	644
Birmingham, England	77.0			1	0.4	3.2	780	703
Elyria, Ohio	69.4			0	0.5		703	634
Grand Rapids, Mich.	63.5	2.4		5	0.14	3.4	651	587
Halle, Germany	72.9			0	0.6	1.6	739	666
Milwaukee, Wis.	67.5		0.6	0	0.2	1.7	686	618
Peoria, Ill.	67.5					4.7	684	616
Plainfield, N. J.	65.8			30.		3.6	667	601
Springfield, Ill.	64.5	1.7		31.		3.2	660	594
Toronto, Ontario, North Toronto Plant	58.5	3.7		28.0	1.8	8.0	605	544

[a] Gas analysis from "American Sewerage Practice" by Metcalf and Eddy, vol. III, Disposal of Sewage, 3d ed., p. 368, McGraw-Hill Book Company, Inc., New York, 1935.
Heating values of gases computed by means of the following data:

Gas	Btu gross	Btu net
Methane (CH_4)	1013.2	913.1
Hydrogen (H_2)	325.0	275.0
Carbon monoxide (CO)	321.8	321.8

In general, gas production will average 0.8 cu ft per person per day. Where household and hotel garbage is ground and dumped into the sewers or where strong industrial wastes are discharged into the sewers, the gas production may average over 2 cu ft per capita per day. At Springfield, Ill., it was determined

that the wastes from 480 people would develop 1 bhp continuously in a gas engine. This community of 72,000 population has a 150-hp gas engine operating constantly at its sewage plant with sewage gas for fuel. The production of sewage gas by the disposal plant at Toledo, Ohio, for a 5-year period was as follows:

Year	Sewage pumped, million gal	Total gas produced, cu ft	1,000 cu ft per million gal sewage
1933	9,357.7	67,700	7.2
1934	8,465.2	91,300	10.8
1935	8,416.2	115,200	13.6
1936	9,513.3	110,400	10.8
1937	10,927.0	92,600	8.5

93. Cleaning Sewage Gas.—One of the difficulties encountered in the use of sewage gas is corrosion resulting from hydrogen sulfide in the gas. Small quantities of sewage sludge also find their way into the gas and form objectionable deposits in engines and gas piping. In order to use sewage gas effectively it is necessary to clean it by the use of suitable traps for removing suspended matter and scrubbers for removing hydrogen sulfide.

TABLE 30.—SOLUBILITY OF GASES IN WATER[a]

Water temperature		Carbon dioxide (CO_2)	Hydrogen sulfide (H_2S)	Methane (CH_4)
Deg C	Deg F			
0	32	1.797	4.371	0.05473
5	41	1.450	3.965	0.04889
10	50	1.185	3.586	0.04367
15	59	1.002	3.233	0.03903
20	68	0.901	2.905	0.03499
25	77	0.772	2.604	0.02542

[a] "Smithsonian Physical Tables," 8th rev. ed., 1st reprint, p. 221.

Table gives volume of gas at 0 C and 760 mm pressure which will be absorbed by unit volume of water at atmospheric pressure and temperature.

Consideration has been given in some sewage plants to the possibility of eliminating the carbon dioxide from the sewage gas, thereby leaving practically pure methane. The possibilities for such purification are good when consideration is given to the rela-

tive solubility of methane, carbon dioxide, and hydrogen sulfide in water as shown in Table 30. At a temperature of 68 F, water will absorb over 90 per cent of its own volume in carbon dioxide, 290 per cent of its volume in hydrogen sulfide, and only 3.5 per cent of its volume in methane.

By passing sewage gas through a water bath of the correct temperature and under suitable conditions, it is possible to eliminate largely the carbon dioxide and hydrogen sulfide and leave a gas that approaches the characteristics of natural gas.

Where carbon dioxide and hydrogen sulfide are to be removed from large quantities of sewage gas, the Girbotol process[1] offers an economical and effective means for effecting their removal. It is a simple cyclic process for scrubbing and recovering acid gases from gaseous mixtures. Gases are scrubbed with solutions of organic bases (amines) to remove acid gases such as hydrogen sulfide and carbon dioxide. The absorbed acid gases are then separated from the amine solutions by heating which reactivates the solutions for further use.

94. Other Gas Fuels.—Blast-furnace gas, producer gas, and coal gas have been used for fuel in gas engines. Typical analyses of these fuels which have been used in engines are contained in Table 31.

95. General Gas Laws.—The volume of a given weight of gas varies with changes in pressure and temperature. In order to make proper corrections for this variation, it is necessary to know the laws under which it takes place.

Boyle found by experiment that *at a constant temperature the pressure of a given weight of gas was inversely proportional to its volume.* Boyle's law can be expressed in the form of a mathematical equation, as follows:

$$\frac{p_1}{p_2} = \frac{V_2}{V_1} \tag{20}$$

or

$$p_1V_1 = p_2V_2 = \text{a constant}$$

where p_1 = absolute pressure of unit weight of gas of volume V_1.
p_2 = absolute pressure of unit weight of gas of volume V_2.

[1] STORRS, B. D., and R. M. REED, The Application of the Girbotol Process to Industry, *Trans. A.S.M.E.*, vol. 64, No. 4, p. 299, May, 1942.

TABLE 31.—COMPOSITION OF MANUFACTURED GAS[a]

Gas	Methane (CH_4), per cent	Ethylene (C_2H_4), per cent	Carbon monoxide (CO), per cent	Carbon dioxide (CO_2), per cent	Hydrogen (H_2), per cent	Oxygen (O_2), per cent	Nitrogen (N_2), per cent	Btu per cubic foot at 60 F 30 in. saturated with H_2O		Cubic foot air required per cubic foot gas
								High	Low	
Coal gas	34.0	6.6	9.0	1.1	47.0	...	2.3	634	560	5.50
Coke-oven gas	28.5	2.9	5.1	1.4	57.4	0.5	4.2	536	476	4.65
Coke-oven gas	33.9	5.2	6.1	2.6	47.9	0.6	3.7	600	538	5.28
Blue water gas	....	....	43.4	3.5	51.8	...	1.3	310	285	2.28
Carbureted water gas	14.8	12.8	33.9	1.5	35.2	...	1.8	578	529	4.85
Oil gas	27.0	2.7	10.6	2.8	53.5	...	3.4	516	461	4.25
Producer gas	2.6	0.4	22.0	5.7	10.5	...	58.8	136	128	1.08
Blast-furnace gas	....	....	26.2	13.0	3.2	...	57.6	93	91.6	0.70

[a] From HASLAM, ROBERT T., and ROBERT P. RUSSELL, "Fuels and Their Combustion," p. 282, McGraw-Hill Book Company, Inc., New York, 1926.

Charles found by experiment that *if the volume of unit weight of gas remains constant, the pressure of the gas is proportional to the absolute temperature.* Charles's law can likewise be expressed in mathematical form as follows:

$$\frac{p_1}{p_2} = \frac{T_1}{T_2}$$

where T_1 = absolute temperature of unit weight of gas at pressure p_1.
T_2 = absolute temperature of unit weight of gas at pressure p_2.

Gay-Lussac found that *for a given weight of gas maintained at constant pressure, the volume is proportional to the absolute temperature.* In terms of a mathematical equation, this law becomes

$$\frac{V_1}{V_2} = \frac{T_1}{T_2} \tag{21}$$

When pressure, temperature, and volume of a given weight of gas all vary, an equation can be developed to take into account all variables. If we consider that

V_1 = volume of unit weight of gas at absolute temperature T_1 and absolute pressure p_1.
V_2 = volume of unit weight of gas at absolute temperature T_2 and absolute pressure p_2.
V' = volume of unit weight of gas at absolute temperature T_1 and absolute pressure p_2.

then from Eq. (20) for Boyle's law at constant temperature we can write

$$\frac{V'}{V_1} = \frac{p_1}{p_2}$$

and

$$V' = \frac{p_1 V_1}{p_2} \tag{22}$$

From Eq. (21) for Gay-Lussac's law at constant pressure we can write

$$\frac{V'}{V_2} = \frac{T_1}{T_2}$$

and

$$V' = \frac{V_2T_1}{T_2} \tag{23}$$

Equations (22) and (23) are both values for V', so we can set them equal to each other and obtain

$$V' = \frac{p_1V_1}{p_2} = \frac{V_2T_1}{T_2}$$

or

$$\frac{p_1V_1}{T_1} = \frac{p_2V_2}{T_2} \tag{24}$$

Equation (24) is the mathematical expression for the general gas law. Examination of this equation shows that for any set of conditions

$$\frac{pV}{T} = \text{constant}$$

Standard gas conditions to which fuel-gas consumption of internal-combustion engines are referred are dry gas at a pressure

TABLE 32.—CONVERSION FROM INCHES MERCURY TO POUNDS PER SQUARE INCH[a]

Inches Hg	Pressure, psia
1	0.491
2	0.982
3	1.474
4	1.965
5	2.456
6	2.947
7	3.438
8	3.929
9	4.421
10	4.912
20	9.823
30	14.735

[a] Based upon "Smithsonian Physical Tables," 8th rev. ed., 1st reprint, p. 180.

of 29.92 in. Hg (at 32 F) and a temperature of 68 F (20 C). The general gas law, Eq. (24), forms the basis for converting total consumed gas volume at the pressure and temperature existing during the volume measurements to standard gas conditions. The equation for this conversion is as follows:

$$V_s = V_m \left[\frac{(p_m - p_n)}{29.92}\right]\left[\frac{527.6}{t_m + 459.6}\right]$$

Table 33.—Combus

Substance	Formula	Molecular weight[a]	Lb per cu ft[b]	Cu ft per lb[b]	Sp. gr. air = 1.000[b]	Heat of combustion[c] Btu per cu ft	
						Gross	Net[d]
Carbon	C	12.01					
Hydrogen	H_2	2.016	0.005327	187.723	0.06959	325.0	275.0
Oxygen	O_2	32.000	0.08461	11.819	1.1053		
Nitrogen (atmos.)	N_2	28.016	0.07439[e]	13.443[e]	0.9718[e]		
Carbon monoxide	CO	28.01	0.07404	13.506	0.9672	321.8	321.8
Carbon dioxide	CO_2	44.01	0.1170	8.548	1.5282		
Paraffin series, C_nH_{2n+2}:							
Methane	CH_4	16.041	0.04243	23.565	0.5543	1013.2	913.1
Ethane	C_2H_6	30.067	0.08029[e]	12.455[e]	1.04882[e]	1792	1641
Propane	C_3H_8	44.092	0.1196[e]	8.365[e]	1.5617[e]	2590	2385
n-Butane	C_4H_{10}	58.118	0.1582[e]	6.321[e]	2.06654[e]	3370	3113
Isobutane	C_4H_{10}	58.118	0.1582[e]	6.321[e]	2.06654[e]	3363	3105
n-Pentane	C_5H_{12}	72.144	0.1904[e]	5.252[e]	2.4872[e]	4016	3709
Isopentane	C_5H_{12}	72.144	0.1904[e]	5.252[e]	2.4872[e]	4008	3716
Neopentane	C_5H_{12}	72.144	0.1904[e]	5.252[e]	2.4872[e]	3993	3693
n-Hexane			0.2274[e]	4.398[e]	.9704[e]	4762	4412
Olefin series, C_nH_{2n}:							
Ethylene	C_2H_4	28.051	0.07456	13.412	0.9740	1613.8	1513.2
Propylene	C_3H_6	42.077	0.1110[e]	9.007[e]	1.4504[e]	2336	2186
n-Butene (butylene)	C_4H_8	56.102	0.1480[e]	6.756[e]	1.9336[e]	3084	2885
Isobutene	C_4H_8	56.102	0.1480[e]	6.756[e]	1.9336[e]		2869
n-Pentene	C_5H_{10}	70.128	0.1852[e]	5.400[e]	2.4190[e]		3586
Aromatic series, C_nH_{2n-6}							
	C_6H_6	78.107	0.2060[e]	4.852[e]	2.6920[e]	3751	3601
Toluene	C_7H_8	92.132	0.2431[e]	4.113[e]	3.1760[e]	4484	4284
Xylene	C_8H_{10}	106.158	0.2803[e]	3.567[e]	3.6618[e]		
Acetylene	C_2H_2		0.06971	14.344	0.9107	1499	1448
Naphthalene	$C_{10}H_8$	128.162	0.3384[e]	2.955[e]	4.4208[e]	5854[f]	5654[f]
Methyl alcohol	CH_3OH	32.041	0.0846[e]	11.820[e]	1.1052[e]	867.9	768.0
Ethyl alcohol	C_2H_5OH	46.067	0.1216[e]	8.221[e]	1.5890[e]	1600.3	1450.5
Ammonia	NH_3	17.031	0.0456[e]	21.914[e]	0.5961[e]	441.1	365.1
Sulfur		32.06					
Hydrogen sulfide	H_2S	34.076	0.09109[e]	10.979[e]	1.1898[e]	647	
Sulfur dioxide	SO_2	64.06	0.1733	5.770	.264		
Water vapor	H_2O	18.016	0.04758[e]	21.017[e]	0.6215[e]		
Air		28.9	0.07655	13.063	1.0000		

Table from "Fuel-flue Gases," American Gas Association, New York, 1940.

All gas volumes corrected to 60 F and 30 in. Hg dry. For gases saturated with water at 60 F, 1.73 per cent of the Btu value must be deducted.

[a] Calculated from atomic weights given in *Jour. Am. Chem. Soc.*, February, 1937.

[b] Densities calculated from values given in grams per liter at 0 C and 760 mm in the International Critical Tables allowing for the known deviations from the gas laws. Where the coefficient of expansion was not available, the assumed value was taken as 0.0037 per degree C. Compare this with 0.003662 which is the coefficient for a perfect gas. Where no densities were available the volume of the mol was taken as 22.4115 liters.

[c] Converted to mean Btu per pound (1/180 of the heat per pound of water from 32 to 212 F) from data by Frederick D. Rossini, National Bureau of Standards, letter of Apr. 10, 1937, except as noted.

TION CONSTANTS

Heat of combustion[c]		Cu ft per cu ft of combustible						Lb per lb of combustible						Experimental error in heat of combustion
Btu per lb		Required for combustion			Flue products			Required for combustion			Flue products			per cent
	Net[d]	O_2	N_2	Air	CO_2	H_2O	N_2		N_2	Air	CO_2	H_2O		
14,093[g]	14,093[g]							2.664	8.863	11.527	3.664			0.012
61,100	51,623	0.5		2.382		1.0	1.882	7.937	26.407	34.344		8.937	26.407	0.015
4347	4347	0.5	1.882	2.382	1.0		1.882	0.571	1.900	2.471	1.571		1.900	0.045
23,879	21,520	2.0	7.528	.528		2.0	7.528	3 990	13.275	17.265	2.744	2.246	13.275	0.033
22,320	20,432	3.5	13.175	16.675		3.0	13.175	3 725	12.394	16.119	2.927	1 798	12.394	0.030
21,661	19,944	5.0	18.821	23.821		4.0	18.821	3 629	12.074	15 703	2.994	1.634	12.074	0.023
21,308	19,680	6.5	24.467	30.967		5.0	24.467	3 579	11.908	15 487	3.029	550	11.908	0.022
21,257	19,629	6.5	24.467	30.967		5.0	24.467	3 579	11.908	15 487	3.029	550	11.908	0.019
21,091	19,517	8.0	30.114	38.114		6.0	30.114	3 548	11.805	15 353	3.050	1.498	11.805	0.025
21,052	19,478	8.0	30.114	38.114		6.0	30.114	3 548	11.805	15 353	3.050	1.498	11.805	0.071
20,970	19,396	8.0	30.114	38.114		6.0	30.114	3 548	11.805	15 353	3.050	1.498	11.805	0.11
20,940	19,403	9.5	35.760	45.260		7.0	35.760	3 528	11.738	15.266	3.064	1 464	11.738	0.05
21,644	20,295	3.0	293	14.293		2.0	11.293	3.422	11.385	14.807	3.138	1.285	11.385	0.021
21,041	19,691	4.5	939	21.439		3.0	16.939	3.422	11.385	14.807	3.138	1.285	11.385	0.031
20,840	19,496	6.0	585	28.585		4.0	22.585	3.422	11.385	14.807	3.138	1.285	11.385	0.031
20,730	19,382	6.0	585	28.585		4.0	22.585	3.422	11.385	14.807	3.138	1.285	11.385	0.031
20,712	19,363	7.5	232	35.732		5.0	28.232	3.422	11.385	14.807	3.138	1.285	11.385	0.037
18,210	17,480	7.5	28.232	35.732		3.0	28.232	3.073	10.224	13.297	3.381	0.692	10.224	0.12
18,440	17,620	9.0	33.878	42.878		4.0	33.878	3.126	10.401	13.527	3.344	0.782	10.401	0.21
18,650	17,760	10.5	39.524	50.024		5.0	39.524	3.165	10.530	13.695	3.317	0.849	10.530	0.36
21,500	20,776	2.5	9.411	11.911	2.0	1.0	9.411	3.073	10.224	13.297	3.381	0.692	10.224	0.16
17,298[f]	16,708[f]	12.0	45.170	57.170	10.0	4.0	45.170	2.996	9.968	12.964	3.434	0.562		[f]
10,259	9078	1.5	5.646	7.146		2.0	5.646	1.498	4.984	6.482	1.374	1.125	4.984	0.027
13,161	11,929	3.0	11.293	14.293		3.0	11.293	2.084	6.934	9.018	1.922	1.170	6.934	0.030
	8001	0.75	2.	3.573		1.5	3.323	1.409	4.688	6.097		1.587	5.511	0.088
									3.287	4.285	1.		3.287	0.071
7100	6545	1.5	5.646	7.146		1.0	5.646	1.409		6.097	1.880	0.529		0.30

[d] Deduction from gross to net heating value determined by deducting 18,919 Btu per pound mol of water in the products of combustion. Osborne, Stimson, and Ginnings, *Mech. Eng.*, p. 163, March, 1935, and Osborne, Stimson, and Fiock, *Bur. Standards Research Paper* 209.

[e] Denotes that either the density or the coefficient of expansion has been assumed. Some of the materials cannot exist as gases at 60 F and 30 in. Hg pressure, in which case the values are theoretical ones given for ease of calculation of gas problems. Under the actual concentrations in which these materials are present their partial pressure is low enough to keep them as gases.

[f] From "Combustion," 3d ed.

[g] *Bur. Standards, Research Paper* 1141.

where V_s = net volume of gas supplied corrected to standard gas conditions.
V_m = total volume of gas measured at p_m and t_m.
p_m = absolute pressure of gas at meter, inches Hg.
p_n = water vapor pressure determined from standard vapor-pressure chart, inches Hg.
t_m = temperature of gas at meter (F).

96. Heating Value.—The heating value of gas is usually determined experimentally in a calorimeter and is expressed as Btu per cubic foot of gas at a specific pressure and temperature. In this experimental determination of the heating value, the hydrogen is burned to water which in turn is condensed to room temperature. When the gas is burned in an engine cylinder, the water formed in the combustion process is not condensed but leaves through the exhaust at the exhaust temperature as a gas or vapor. Thus the heat of vaporization of the water formed cannot be recovered as useful work in the engine. This has created a dual classification of heating values. The experimentally determined value with the water condensed to a liquid is called the *gross heat content*, or *high-heat value*, while this minus the latent heat of vaporization of the water formed from the hydrogen content of the fuel is called the *net*, or *low-heat, value*.

The difference between the gross and net heating values, or hydrogen loss as it is often referred to in combustion calculations, for the more common constituents of natural gases, is as follows:

Gas	Per cent total heat content		
	Gross	Net	Hydrogen loss
Methane (CH_4)..	100	90.12	9.88
Ethane (C_2H_6)...	100	91.57	8.43
Propane (C_3H_8)..	100	92.08	7.92
n-Butane (C_4H_{10})	100	92.37	7.63

The heating values, both gross and net, can be calculated readily from a knowledge of the chemical composition of the gas and with the aid of the data on heat of combustion in Table 33. Consider a natural gas having a composition as follows:

Gas	Per Cent of Volume
Methane	74
Ethane	18
Carbon dioxide	1
Nitrogen	7
	100

At 60 F and 30 in. Hg this gas in a dry condition would have a gross heating value as follows:

		Btu per Cu Ft
Methane	0.74 × 1,013.2 =	750
Ethane	0.18 × 1,792 =	323
Gross heating value	=	1,073

and a net heating value calculated as follows:

		Btu per Cu Ft
Methane	0.74 × 913.1 =	676
Ethane	0.18 × 1641 =	295
Net heating value	=	971

In the illustrations given, the carbon dioxide and nitrogen add nothing in the combustion process, and they can, therefore, be ignored insofar as determining the heating value of the fuel is concerned.

97. Flow of Gas in Pipes.—Many equations have been developed for the flow of gas in pipes, some dealing only with gas flow at low pressures where only a slight difference of pressure exists between the points of origin and delivery of the gas, while others deal with gas when a considerable pressure drop occurs between origin and delivery points. The equation developed by Weymouth is used very extensively for natural-gas work and is as follows:

$$Q = 18.06 \frac{T_0}{p_0} \sqrt{\frac{(p_1^2 - p_2^2)d^{5.33}}{GTL}}$$

where Q = quantity of gas flow, cubic feet per hour at $T_0 p_0$.
d = internal diameter of pipe, inches.
L = length of line, miles.
G = specific gravity of gas referred to air.
T_0 = absolute temperature, deg F at which gas measured.

T = absolute temperature, deg F at which gas is flowing in pipe.
p_0 = absolute pressure, psi at which gas measured.
p_1 = absolute pressure, psi at point of origin.
p_2 = absolute pressure, psi at point of delivery.

Figure 79 gives the head loss of gas flowing in various sizes of pipes with the specific gravity of the gas taken as 0.65. For

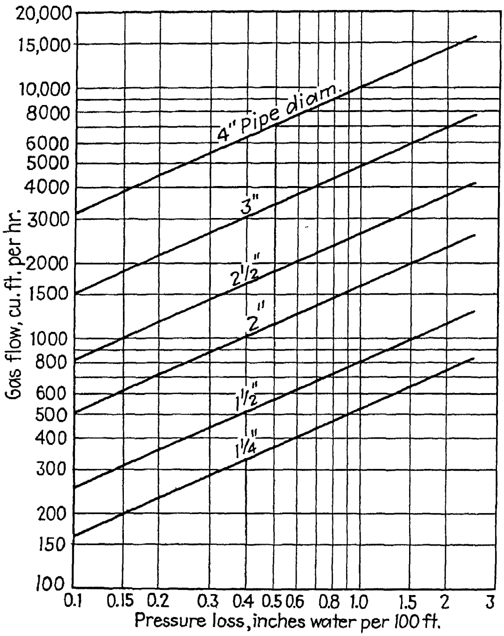

FIG. 79.—Pressure loss in gas lines for gas having a specific gravity of 0.65.

any other specific gravity, a correction factor

where K = correction coefficient.
G = actual specific gravity of gas.

can be applied to results obtained from this chart. Thus for specific gravities less than 0.65 the head loss for a given flow

of gas will be less than the values shown on the chart, while for specific gravities greater than 0.65, the head loss will be greater.

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CHAPTER XI

LUBRICATION

We are accustomed to thinking of lubricating oil as that product derived from petroleum which permits one surface to slide over another with less friction and wearing of the two surfaces than if it were not used. The problem of correct lubrication of internal-combustion engines, however, is extremely complex, involving the selection of the proper lubricant for the particular engine and the conditioning of that oil in order to obtain the maximum service life from it.

98. Lubrication Requirements.—The lubrication of an internal-combustion engine is not to be compared with that of anything else mechanical. Although the internal-combustion engine combines the forms and characteristics of many other mechanical devices, its needs are entirely its own, and these needs dictate the lubrication requirements of the engine. As has been pointed out so clearly,[1]

> It is the only device conceived by man that discharges the "ashes" of its combustion into its lubricant, yet depends upon the resulting mixture for both lubrication and cooling; that surrounds the heat of combustion with a film of oil which, in the presence of this heat, must lubricate, cool, seal, and scavenge, yet resist complete destruction; that inhales abrasives, deposits them in the oil, creating an abrasive mixture, which when reduced to the thin film of lubrication, damages the parts the oil is intended to protect; that confines within its own compact mass both the source of its power and the cause of its eventual destruction.

Progress in the design and fabrication of internal-combustion engines involving higher rotative speeds, higher piston speeds, and increased mean effective pressures has increased the problems confronting those producing lubricants for this service. In fact the

[1] "The Fundamentals of Automotive Engine Lubrication," vol. 1, No. 5, p. 6, Shell Oil Company, Inc.

successful operation of any internal-combustion engine is so greatly influenced by correct lubrication that the lubricant can be rightfully called the "life blood" of the engine.

Some of the duties imposed upon the lubricants used in internal-combustion engines are as follows:

1. Prevent wear of cylinders, rings, pistons, and bearings.
2. Prevent ring sticking.
3. Not corrode bearing metals.
4. Seal rings against blow by.
5. Not carbonize on underside of pistons.
6. Keep bearings and pistons cool.
7. Keep engine interior and pistons clean.
8. Not deposit sludge on oil filter or in oil lines or oil passages.
9. Form no deposits in oil cooler.
10. Form no varnish on exposed metal surfaces.
11. Form no carbon in exhaust ports.
12. Lubricate air compressors, gears, chains, and other engine mechanisms.
13. Separate readily from water.
14. Provide easy starting at low temperatures.
15. Lubricate for long periods without changing.
16. Recondition properly.
17. Maintain low oil consumption.

In addition, the operator frequently expects that the lubricating oil will alleviate or entirely prevent ring scuffing, piston seizure, cylinder scoring, bearing cracking, contamination, and valve sticking as well as fuel dilution resulting from poor combustion.

No oil can fulfill all these requirements, and fortunately no single installation requires that the lubricant perform all these duties. As a consequence, it is necessary to select that lubricant which will best meet the conditions presented by the particular installation in question.

99. Quality of Lubrication.—While the plant operator buys lubricating oil for an internal-combustion engine, what he is really buying is a *function* performed by that oil which we term *lubrication*. As a result, it is necessary to evaluate the lubricant used in terms of its ability to perform the services for which it is intended. If under a particular set of service conditions a given volume of oil or grease gives correct lubrication for a longer time than others,

its quality is, for those conditions, better than the others. After all, this is the kind of quality a user wants.

Apart from the lubricating ability of an oil, but vitally necessary to obtaining the greatest functional use of the lubricant, is the correct application of that oil. The best oil obtainable is of no use unless it is applied correctly.

100. Source of Lubricating Oils.[1]—Lubricating oils are derived from the portions of petroleum, usually called distilled and residual lubricating oil fractions, remaining after the gasoline, kerosene, and light fuel oils have been removed. In the making of lubricating oils, the wax or petrolatum is first removed, after which the lubricating oil fractions are either refined with chemicals, or the undesirable elements are removed with solvents. Following this the lubricating oil is filtered through fuller's earth.

Crude petroleums in the United States are commonly described as being *paraffin-base* oils, *naphthene-base* oils, or *mixed-base* oils, the latter displaying properties intermediate between the other two. Oils from the eastern United States are paraffin-base oils; those from the Gulf Coast and California areas are naphthene-base oils; and the mid-continent fields produce the mixed-base oils. In reality the division of crudes into three grades is incomplete, but this classification has been used extensively by the petroleum industry in the United States and will probably continue to be used for general reference purposes. In reality, all crude oils are complex mixtures of various series of hydrocarbons, and these terms should not be taken to mean that any crude oil consists wholly or even predominately of one family of hydrocarbons.

Lubricating oils are often classified in accordance with the base crude used in their compounding. Thus lubricants made from eastern crudes are spoken of as *paraffin-base lubricating oils.* Likewise, lubricants made from Gulf Coast and California crudes are termed *naphthene-base lubricating oils*, and those compounded from mid-continent crudes are termed *mixed-base lubricating oils.*

In the early development of lubricating-oil refining processes, it was found that the best lubricating oils were produced from the paraffin-base crudes. The advent of solvent refining, however,

[1] For a more extensive discussion see Clower, James I., "Lubricants and Lubrication," p. 21, McGraw-Hill Book Company, Inc., New York, 1939.

placed methods in the hands of the oil refiner for making from naphthene-base and mixed-base crudes lubricating oils that were as good as the lubricants produced from the eastern oils.

Straight mineral oils are those lubricants produced by the acid-refining process. Solvent-refined oils usually contain additives, since it is often necessary to provide an oxidization inhibitor in such lubricants.

101. Selection of Lubricating Oil.—The refining and compounding technique now employed permits the refiner to produce the many types of lubricating oils required for an extremely wide variety of lubricating requirements. When it is realized that lubricants derived from crude petroleum are used in wrist watches no larger than a quarter as well as in the gear trains of heavy machinery where extremely high pressures are encountered, the magnitude of the problem confronting the lubricant manufacturer is better appreciated. This wide divergence of requirements, in both pressure and temperature under which the oils must operate successfully, necessitates the production of many special lubricants suitable only in specific operations. With so many types of lubricants available, it is necessary that the engine operator exercise special care to select the correct lubricants for his engines. The test of the suitability of any oil is the performance of that lubricant under actual operating conditions.

It is impossible for the plant operator to be familiar with the advantages and limitations of each individual lubricant produced. He must rely to a large degree upon the services of lubrication specialists to guide him in the selection of suitable lubricating oils. While the services of the lubrication specialist should be utilized in the selection of the proper lubricating oil for a particular engine, there are certain basic fundamentals regarding lubrication and lubricating oils with which it is advisable for the plant operator as well as the plant designer to be familiar. At the outset it should be emphasized that in selecting a lubricating oil more reliance should be placed in the reputation of the oil refiner and the actual field performance of a particular lubricant than on laboratory tests. Furthermore, once a suitable oil has been selected for use in a particular engine, continue the use of that oil. Experimentation with different types of lubricants may cause difficulty in the operation of the engine, or may even result in a large repair bill through improper or inadequate lubrication.

The tests which are usually made on lubricants can be classified as chemical, physical, and mechanical. It is not within the scope of this chapter to discuss all of them in detail. Tests made will be listed together with references as to where detailed information regarding them may be obtained. Only those tests of special interest from the viewpoint of the lubrication of internal-combustion engines will be discussed.

102. Lubricating-oil Tests.[1]—The following is a summary of the tests usually made on lubricating oils to determine their characteristics and qualities.

Chemical tests:
1. Acidity or alkalinity.
2. Oxidation.
3. Precipitation number.
4. Saponification number.
5. Sulfur content.
6. Carbon residue.
7. Corrosion.

Physical tests:
1. Cloud and pour points.
2. Color tests.
3. Flash and fire points.
4. Dilution by fuel oil.
5. Emulsification.
6. Evaporation.
7. Viscosity.
8. Gravity.
9. Water and sediment.

Mechanical tests:

Mechanical tests are made on wearing machines to determine the lubricating value of the oil.

103. Viscosity.[2]—The viscosity of a lubricating oil must be given careful consideration in the selection of the proper oil for any internal-combustion engine installation. One of the primary requirements of a satisfactory lubricating oil is a suitable viscosity-temperature characteristic. The greater the temperature range through which the oil must operate, the less should be the relative change in the viscosity of the oil. If the viscosity of the

[1] See Clower, *op. cit.*, also standards of the American Society of Testing Materials for standard test procedures.

[2] See Art. 79, Chap. IX, for discussion of viscosity.

oil is extremely high when the oil temperature is low upon starting the engine, oil flow will be restricted and lubrication will be poor. After the engine and oil have warmed up, the viscosity of the oil must be sufficient to maintain a film thickness that will prevent metal-to-metal contact and resulting excessive wear. While a high-viscosity oil will result in a lower oil consumption, nevertheless considerable power loss results from the increased friction caused by the higher viscosity. In view of the conditions and limitations imposed by the oil viscosity, it is always desirable to use the lowest viscosity oil that will maintain sufficient oil film to prevent metal-to-metal contact at the engine operating temperatures.

104. Oxidation.—Where lubricating oil is used over and over as it is in an internal-combustion engine, it should be relatively free from oxidation difficulties. Admittedly, lubricating oils subjected to the heat of combustion in the cylinder will burn. The major portion of the lubricating oil, however, is not subjected to this intense heat and should not oxidize appreciably at the temperatures encountered in the engine crankcase and in the piston when used for piston cooling.

105. Carbon Residue.—While Conradson carbon or other carbon-determination tests are often called for in specifying lubricating oil, their value is open to question.[1] This authority states:

> However, although it is true that (carbon) tests may be of value in comparing oils of similar origin which have received a similar treatment, they lose their practical significance when comparisons are made of oils coming from different sources or subjected to different treating procedures.
>
> This is due to the difference in the characteristics of the carbon formed in the engine. Some types of carbon are rather fluffy in character and thus easily eliminated through the exhaust while others form a hard film on pistons or on cylinder walls. It is also debatable whether an oil with an exceptionally low carbon residue or coke number is desirable, since a slight carbonaceous deposit may be of some value in protecting the oil from direct action of the metal. These functions are, however, not yet definitely settled.

[1] Kalichevsky, Vladimir A., "Modern Methods of Refining Lubricating Oils," p. 16, American Chemical Society *Monograph* No. 76, Reinhold Publishing Corporation, New York, 1938.

106. Addition Agents.—Since lubricating oils are used for widely varying types of service in our industrial world today, it is only natural that for certain service conditions it has been found necessary to add various compounds to mineral lubricating oils in order that the lubricant will meet the special requirements desired. These "additives" are generally put into lubricating oils to act as pour-point depressants, oiliness carriers and extreme pressure materials, viscosity-index improvers, oxidization inhibitors, and color improvers. The services intended of the various additives are self-explanatory from the foregoing enumeration. Of these additives, probably oiliness carriers, oxidization inhibitors, and color improvers are of most interest to those concerned with internal-combustion engines.

Oiliness carriers are generally vegetable or animal oils which are added for the purpose of maintaining an oil film between adjacent moving metal surfaces. There is a tendency with some oiliness carriers toward affecting the oxidization stability of the oil adversely. Oxidization inhibitors, the use of which is still in the experimental stage, apparently work best with highly refined oils.

Color improvers have been developed to give lubricating oils refined from mid-continent and California crudes the green fluorescence of the Pennsylvania oils. In this connection Kalichevsky[1] points out that

> In service oils tend to develop black carbonaceous material which remains at least partially suspended in the oil. If the oil is fluorescent and is examined in reflected light, the presence of these black particles is concealed and the oil continues to appear relatively little altered. When the nonfluorescent oils are examined in transmitted light all these black particles are distinctly visible, and the black appearance of the oil leaves the customer with the impression that the oil has deteriorated rapidly in service. Actually, however, the fluorescent oil may contain considerably larger quantities of carbonaceous materials and may have deteriorated to a much larger extent than the nonfluorescent oil rejected by such visual inspection.

In addition to those additives which may be employed by the oil refiner in compounding lubricating oils to meet special requirements, the diesel-plant operator and designer have for the past several years been bombarded with requests to try numerous

[1] *Ibid.*, p. 196.

additives or "dopes" which are to be mixed directly in the fuel or lubricating oil. Like the elixirs of the alchemists, these dopes were supposed to cure every known ill of poor design or faulty operation. Unfortunately, there is just enough fact to substantiate a part of the claims made for these dopes. When such materials are added directly to the lubricating oil of an engine, they should be used with caution since in rectifying one difficulty they may cause others more serious than the trouble rectified.

107. Methods of Engine Lubrication.—The method of lubrication employed in any engine is determined by its design. In general, however, it can be said that all slow-speed engines use similar methods of lubrication in which two independent lubrication systems are employed, while the small high-speed engines use only a single lubricating system.

The large slow-speed units usually employ two independent lubricating systems consisting of a full-pressure force-feed system for lubricating the main, crankpin, and wristpin bearings, camshafts, gears, and other rotating parts, and a separate force-feed system for lubricating cylinder walls. It is the usual practice to use a type of oil in each system which will best suit the conditions and limitations of the lubricating problem in each. Thus the oil used for the cylinders is often of a higher viscosity than that employed for bearings. Usually new oil is used in the cylinder lubricators.

As lubricating oil is fed to the cylinders, a portion is retained on the cylinder walls, some is burned in the fuel-combustion zone, and some drains off the cylinder walls into a special compartment or into the crankcase where it mingles with the oil for the main lubricating system. This oil which is drained or scraped off the piston skirt or lower portion of the cylinder carries with it carbon, water vapor resulting from fuel combustion, and some cracked residue of the cylinder lubricant. In general, this oil finds its way into the pressure-lubricating system for bearing lubrication. The oil in the bearing-lubricating system is not subjected to the temperatures prevalent in the cylinder, and consequently it is not used up at so rapid a rate as is the cylinder lubricant. This oil, after proper cleaning, is used over and over in the engine.

In most of the higher speed engines, a single lubricating system is employed. The oil is forced through the system by means of a pump, and cylinder walls are lubricated by means of splash

carried up by the lower portion of the piston skirt. Thus in engines of this type the cylinder lubricant, instead of being new oil, is furnished from a mixture of oil which has been in the crankcase and new oil added to compensate for that consumed during operation of the engine.

108. Lubricating-oil Impurities.—Lubricating oil used in an internal-combustion engine is subject to continual contamination while the engine is in operation. Since it acts as a scavenging medium, it is accumulating carbon, water, dirt, metallic chips, as well as other contaminants. It is also subject to chemical change since unsaturated hydrocarbons in the presence of heat, with a metal present to act as a catalytic agent, may be joined to form entirely different compounds. Impurities can also be introduced in the lubricating oil by too frequent handling, particularly in those regions subject to severe dust conditions.

In all practical cases, part of the impurities and products of decomposition of the lubricant and fuel appears in the crankcase oil, and part forms the deposits in the engine. The design, mechanical condition of the engine, and type of operation control the distribution of impurities and the rate of engine fouling to a greater extent than the lubricating-oil characteristics. It is, therefore, impossible to separate the lubricating service from other phases of operation.

Economy dictates that the oil be used in an engine as long as possible. Since the contamination which is constantly being added to the lubrication oil is harmful to the engine when its concentration becomes too great, it is necessary to remove the contaminants from the oil which cause harm to the working parts of the engine. Thus means should be provided for the purification of the oil used if economical service is to be obtained from it.

109. Purifying Lubricating Oil.—Many methods are used for purifying lubricating oil used in internal-combustion engines. Before discussing these several methods in detail, it is advisable to consider certain general features of oil reconditioning. All methods of purification are not applicable to all conditions encountered in the operation of internal-combustion engines, nor are all lubricating oils satisfactorily reconditioned in all types of equipment sold for this purpose.

Basically there are differences in lubricants resulting from the method of refining employed as well as the method of compound-

ing the oil. Improvements in refining technique are continually being made which alter the characteristics of lubricants offered. These improvements in lubricating oils must keep pace with refinements and improvements in the design and construction of engines necessitating higher grade lubricants.[1] Furthermore, the characteristics of a lubricating oil may be changed considerably by some reclaiming processes now used.

All lubricating oils deteriorate when subjected to high temperatures. The safe maximum temperature to which an oil should be heated, if long life is to be obtained from it, is 200 to 225 F. As oils are heated above this value, in general, the oxidization rate doubles for each 20 F increase in the temperature. This fact is borne out by extensive laboratory tests on all types of lubricants employed for internal-combustion engines, steam engines, and steam turbines.

Laboratory tests of thousands of samples of lubricants removed from internal-combustion engines show the further fact that heating the lubricating oil to a high temperature removes little if any of the fuel-oil dilution present in the lubricating oil. This is probably due to the fact that the distillation range of most fuel oils used in diesel engines is practically identical with that of the distillation range of the lighter portions of the lubricating oil.

110. Purifying Methods.—There are three basic methods for purifying lubricating oil, namely, settling, filtering, and chemical reclaiming. In many instances, combinations of these several methods are employed to remove contaminants from lubricating oils. Reconditioning may be done by the batch method where the oil is removed from the engine for treatment and then replaced, or it may be accomplished by continuous reconditioning where some of the lubricating oil is constantly being by-passed through the treating equipment while the engine is operating. It is usually not economical to pass all the oil being circulated in the engine through the reconditioning equipment except in the case of small engines where the rate of flow is low.

111. Settling.—Perhaps the simplest method for removing impurities from lubricating oil is by settling. If the oil is removed from an engine, placed in a tank where it is heated, and left undisturbed for a sufficient length of time, all the carbon, sludge, water,

[1] Lubrication Problems Created by the Modern Diesel, *Lubrication*, The Texas Company, New York, vol. 25, No. 10, October, 1939.

and other contaminants will settle to the bottom of the tank. The clean oil contained in the upper portion of the tank can then be drawn off, replaced in the engine, and used until it again

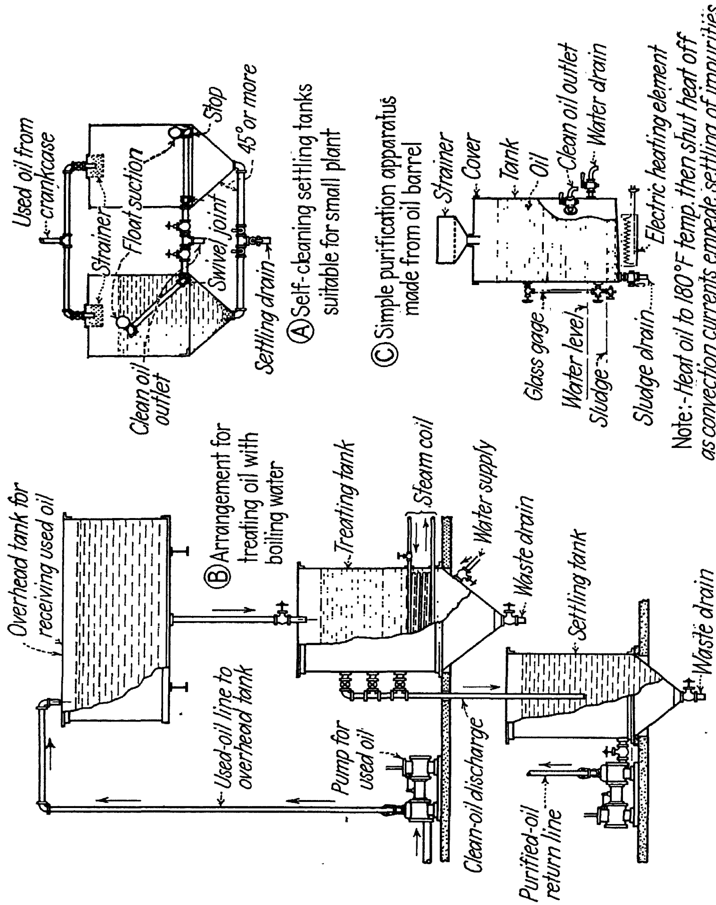
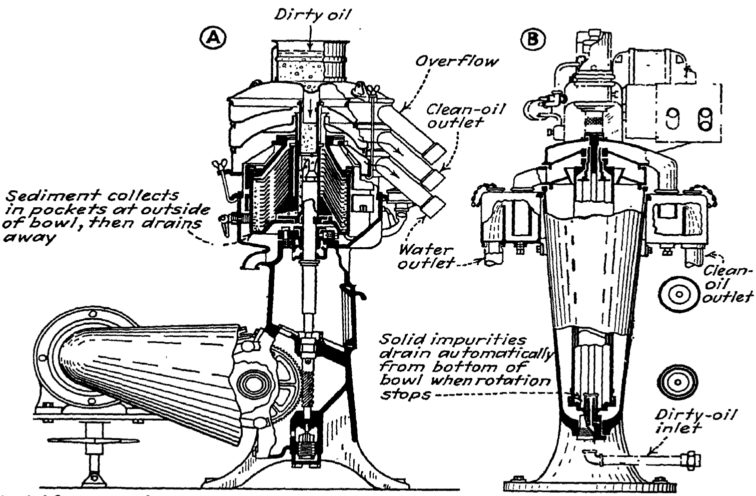


FIG. 80.—Typical arrangements for purification of lubricating oil using batch settling. (*Courtesy of Power.*)

becomes too contaminated for efficient service. Such a treatment method is inexpensive, although it is open to the objection that a large oil storage must be provided when the size of the

engines in the plant exceeds 500 hp. It cannot be used where the settling tanks are subject to disturbance as on ships or railroad locomotives. Batch settling is the only method, generally accepted at present, for reducing the soluble content of lubricating oils.

112. Centrifuging.—A centrifuge employs accelerated precipitation in purifying lubricating oil. The force produced by the



Centrifugal purifier in which oil is divided into thin layers to increase separation rate

Centrifugal purifier with long small-diameter bowl for separation

Fig. 81.—Construction details of centrifuges used for lubricating-oil purification. (*Courtesy of Power.*)

centrifuge on the contaminants in the oil is several thousand times that of gravity, and consequently the rate of separation is very much faster than in a gravity-settling system. This method of purifying effectively removes all solids and liquids heavier than the oil which are not in solution in the oil. In general, the centrifuge is most effective where large amounts of foreign contamination or oil-decomposition products are progressively contaminating the oil during service. Lubricating oil will seldom return to its original color in the centrifuging process. Carbon in colloidal form as low as 0.018 per cent by volume will

make the oil appear black. Such an amount of carbon is harmless in the lubricant.

Satisfactory purification of lubricating oil with a centrifuge is obtained by following these simple rules given by the Texas Company.[1]

1. Set the oil-flow rate at 60 per cent of the maximum capacity of the centrifuge for the particular oil and temperature.
2. Maintain the oil temperature at the centrifuge constant at 180 F.
3. Clean the centrifuge bowl every 6 hr or oftener.
4. Operate the centrifuge continuously.
5. Maintain the original viscosity of the oil.

When a centrifuge is properly maintained and operated intelligently, it will do a very creditable job in purifying the lubricating oil used in an internal-combustion engine. Probably the two most common faults committed by operators in handling centrifuges deal with heating of the oil and cleaning of the centrifuge bowl.

In order to clean up a lubricating oil satisfactorily, it is necessary to heat the oil in order to reduce its viscosity and specific gravity. With the viscosity lowered, particles of dirt and drops of water move more freely through the oil. The lowered specific gravity of the oil results in greater difference in centrifugal force acting upon the oil as compared with that acting upon the water and other foreign matter and, consequently, a more positive separation in the centrifuge. Heating of the oil must be done properly and not carried to high temperatures. It happens that some contaminants, including water, which are insoluble in the oil at low temperatures are highly soluble at increased temperatures. Likewise, some additives used to improve the lubricity of the oil may exhibit a tendency to saponify or undergo other changes at elevated temperatures.

Improper heating of the lubricating oil may result in considerable oxidization of the lubricant, and conditions have occurred where the amount of carbon produced in an electric heater on the centrifuge was greater than the amount of carbon being produced through service of the oil in the engine. In one instance, the elimination of the electric-heater element resulted in a much cleaner oil being delivered by the centrifuge. Some authorities

[1] *Ibid.*

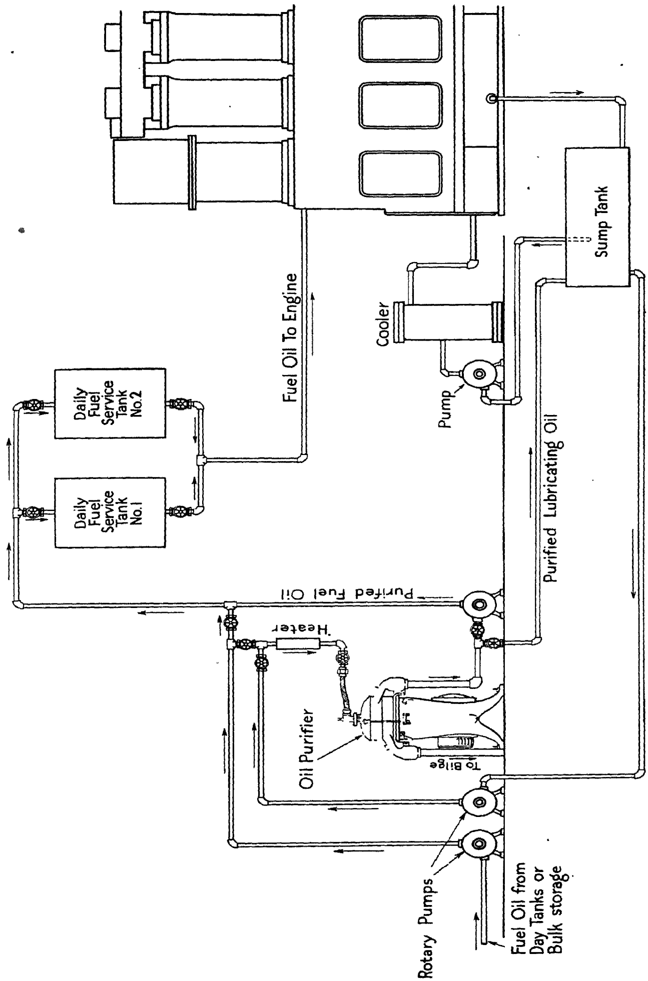


FIG. 82.—Combination fuel and lubricating-oil purification system using a single centrifuge. (*Courtesy of De Laval Separator Company.*)

recommend the elimination of electric strip heaters, and in their place they recommend the use of a steam coil operating at not over 5 lb steam pressure. Such a steam coil will keep the temperature of the oil within safe limits.

The neutralization number of an oil can generally be held within reasonable limits through the introduction of 5 to 8 per cent by weight of mineral-free hot water with the oil as it enters the centrifuge. The water will be scrubbed through the oil, assimilating a portion of the acid present. When the centrifuge separates the water from the oil, extraction of a portion of the acidic materials in the oil is obtained. The water should be as hot as the oil, and preferably 10 to 20 F warmer than the oil. The use of water is somewhat tricky and requires the operator to be somewhat careful in its use.

The centrifuge does not remove additives or detergents employed in the compounding of some types of lubricating oils.

Centrifuge equipment may be used either for intermittent cleaning of batches of oil removed from an engine, Fig. 82, or it may be arranged for continuous operation where a portion of the lubricating oil from one or more engines is continually passing through the centrifuge, Fig. 83. The latter method of operation is much to be preferred. The arrangement of centrifuge equipment in any case is dependent upon the type of lubrication system provided on the engine. In some cases a batch-cleaning arrangement must be used since a continuous system cannot be installed on the engine. In other cases it is found desirable to provide a separate centrifuge for each engine for continuous by-pass cleaning. Some installations use one or more centrifuges for continuous by-pass cleaning of the oil for a group of engines.

113. Filtering.—Filters are used extensively to remove lubricating-oil contaminants. Originally filtering elements were largely confined to simple strainers for removing the larger particles of dirt, metallic chips, and other foreign materials. Developments in the construction of filters have produced many types and styles, although they all may be classified in three groups including (1) strainers, (2) absorbent filters, and (3) adsorbent filters.

Strainers may be composed of wire cloth, cloth bags, or the edge-type unit in which the oil passes through small openings between flat metallic plates. *Absorbent filters* employ cotton waste, multiple layers of cloth, spun-glass fiber, cellulose, and

other materials capable of absorbing the foreign materials from the lubricating oil as it passes through the filtering mediums. *Adsorbent filters* use raw fuller's earth or synthetic materials resembling fuller's earth in their action in adsorbing contaminants from the oil.

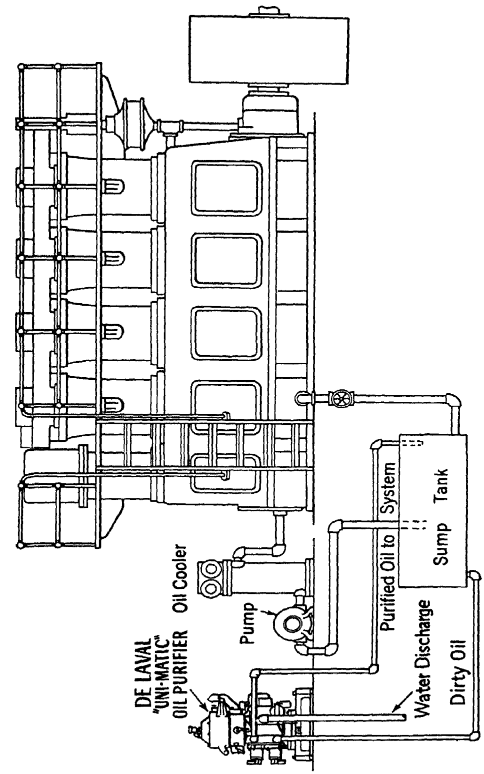


Fig. 83.—Continuous by-pass method of lubricating-oil purification on a pressure-lubricated engine. (*Courtesy of De Laval Separator Company.*)

The absorbent- and adsorbent-type filters are used extensively and have found favor among operators. These filters seem to be best suited for use in those instances where the amount of contamination to be removed from the oil is not great. Where an extremely dirty oil is to be cleaned, the cost of filter-replacement cartridges becomes excessive in comparison to the performance

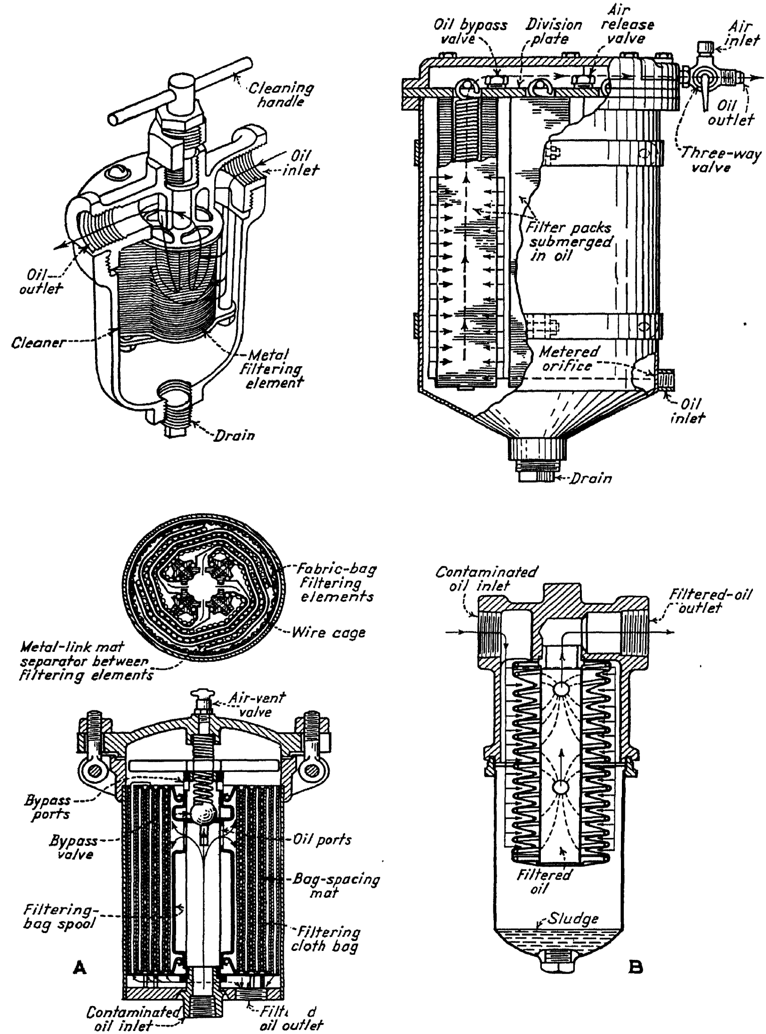


FIG. 84.—Typical fabric, cellulose, and metal-disk filters. (*A*) and (*B*) Fabric filtering elements used. (*C*) Filtering element of metal disks. (*D*) Special paper used for filtering element. (*Courtesy of Power.*)

obtained from a well-operated centrifuge. There seems to be some question as to just how effective filters are in removing

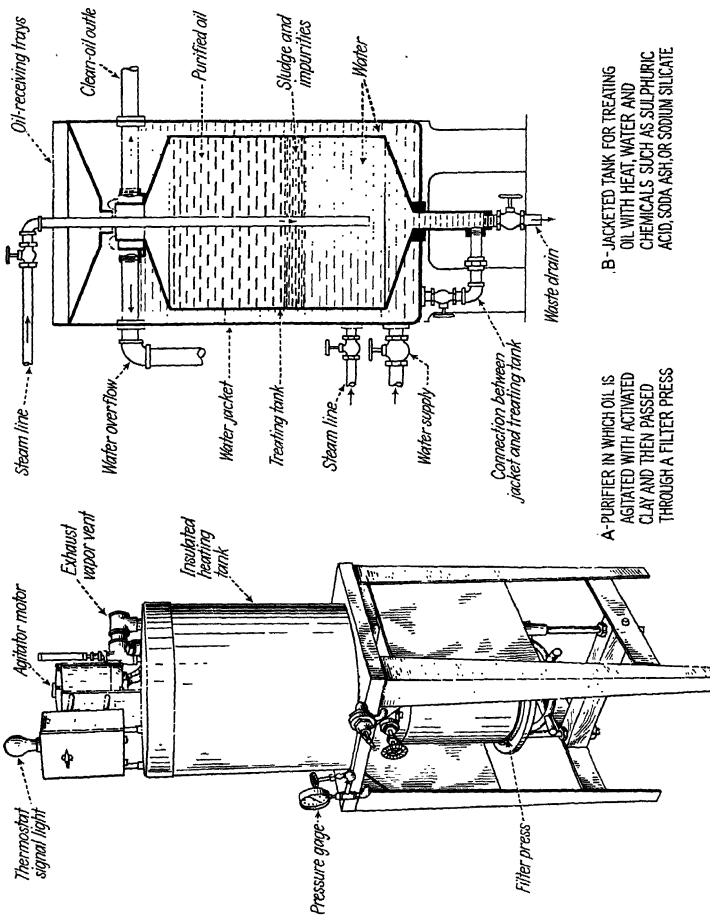


FIG. 85.—Activated clay and chemical reclaimers employed for lubricating-oil purification. (*Courtesy of Power.*)

liquid contaminants from lubricating oil, particularly when these contaminants are in solution in the oil.

While activated-earth filters are being used satisfactorily with oils that contain oxidization inhibitors, many refiners recommend

that lubricants containing both oxidization inhibitors and detergents should be filtered only with cellulose-type filtering mediums such as cellulose and cotton waste. This is because the materials used to obtain detergency characteristics are much more sensitive to removal from the oil by activated earth than most oxidization inhibitors.

114. Chemical Reclaimers.—Several makes of oil-purifying units are being marketed in the United States which are known among plant operators as *chemical reclaimers*, although their action in part is similar to that of the adsorbent-type filter. These units employ high-temperature distillation coupled with filtration through fuller's earth, or high-temperature distillation together with the mixing of fuller's earth with the oil being treated followed by pressure filtering. The successful handling of such equipment calls for considerable care on the part of the operator, and the effects of high temperatures upon the lubricating oil may be extremely detrimental.

There are chemical reclaimers that may be either acceptable or undesirable, depending upon the nature of the lubricating oil treated, method of application of heat, ratio of chemicals to oil, strength of chemicals, and other factors entering the problem of oil reconditioning. It is extremely difficult, therefore, to tell just how successful a particular chemical reclaiming method will be with a certain lubricating oil.

The cost of chemicals and electrical energy used for heating the reclaimer is considerable. There have been instances, particularly among pipe-line operators, where this type of equipment has been ruled out as a means for lubricating-oil conditioning because of its high operating cost as compared with centrifuging or filtering through waste or cellulose materials.

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CHAPTER XII

ENGINE COOLING

When an internal-combustion engine is in operation, it derives its propulsion energy from the burning of oil or gas in each cylinder. The heat energy produced is only partially converted into power at the engine crankshaft. That which is not converted into power is lost through mechanical friction, radiation to the surrounding atmosphere, rejection in the exhaust, and in the water circulated through the engine for cooling the cylinders, lubricating oil, and exhaust lines. It is the quantity of heat dissipated to the cooling water and the means for its disposal which will be discussed in this chapter.

115. Heat Rejected to the Cooling System.—The percentage of the total heat supplied to the engine which finds its way to the cooling water varies considerably for different makes and types and is influenced by the stroke cycle as well as the individual design of the engine. Several independent studies indicate that there is no uniformity of opinion among engine designers as to the quantity of heat per brake-horsepower at full load which is removed by the cooling water. These studies have shown that the heat dissipation varies from 1,500 to 4,500 Btu per bhp-hr, depending upon who furnished information. Most plant designers follow the approximate rule of D.E.M.A.[1] that "an average of 3,000 Btu per bhp-hr is an approximation which may be used for the amount of heat rejected by an engine to the cooling water system."

While actual heat dissipation to the cooling water for a particular engine should be checked with the manufacturer, it is possible, through a knowledge of engine performance, to estimate fairly accurately the amount of heat rejected. In a diesel engine, for example, the practice of the D.E.M.A. since 1935 has been to

[1] "Standard Practices," 1935, p. 71, and "Marine Diesel Engine Standards," 1940, p. 105, issued by the Diesel Engine Manufacturer's Association, New York.

make fuel-oil-consumption guarantees upon the basis of a gross (high) heating value of the fuel oil of 19,000 Btu per lb. If a diesel engine consumes 0.4 lb of fuel oil per brake horsepower-hour at full load, (Fig. 14) of which 2,545 Btu is the actual heat equivalent of 1 bhp-hr delivered at the crankshaft, we can readily obtain the total amount of heat energy rejected by the engine as follows:

	Btu
Heat supplied per bhp-hr (19,000 × 0.4)	7,600
Heat to mechanical work	2,545
Heat rejected by the engine per bhp-hr	5,055

This heat rejected by a nonsupercharged engine is generally divided about equally between that which is absorbed by the cooling water and that which is discarded through the exhaust, radiation, and friction. Thus, where no oil cooling or exhaust cooling is done, the heat absorbed by the cooling water per brake horsepower-hour will be approximately 0.5 × 5,055, or 2,528 Btu. For engines without oil-cooled pistons or exhaust cooling, therefore, it is generally safe to estimate that 2,500 to 2,700 Btu will be absorbed by the cooling water for every brake horsepower-hour at full load. On the other hand, where lubricating-oil cooling is added, another 700 to 900 Btu are absorbed by the cooling water, and the water jacketing of the exhaust manifold will add another 200 or more Btu, depending upon the exhaust temperature. Thus, when lubricating-oil cooling and exhaust cooling are both added, the heat absorbed by the cooling water will range from 3,400 to 3,800 Btu per bhp-hr. Supercharging tends to change this distribution with the exhaust carrying off a larger portion of the waste heat and the cooling water taking as little as 1,800 Btu per bhp-hr at full load. From these figures it is readily apparent that no definite data on heat rejection to the cooling water can be given which are applicable to all engines.

Reference has been previously made to the fact that the amount of heat not converted into mechanical work is rejected by the engine and is divided between that heat which finds its way to the cooling water and that which is dissipated through the exhaust, mechanical friction, and radiation. The amount of heat dissipated as mechanical friction and radiation is a relatively small portion of the total heat supplied to the engine and for any particular engine design is more or less constant regardless of load on

the engine or regardless of the distribution of the heat between the cooling water and exhaust gases. In view of this latter condition, therefore, the greater the percentage of the rejected heat absorbed by the cooling water the less the percentage of heat finding its way out the exhaust.

116. Cooling-water Requirements.—The amount of water required for cooling is determined by the quantity of heat to be removed from the engine or engines in operation and the temperature rise of the water passing through the engines. The quantity of heat removed is measured in British thermal units. A British thermal unit is, for all practical purposes, defined as the quantity of heat required to raise one pound of water one degree Fahrenheit. Thus, if a pound of water is heated 10 F, it absorbs 10 Btu in the process. This relationship between quantity of heat, weight of water, and temperature rise, where the water is kept below the boiling point, is given by means of the following equation:

$$H = W(t_2 - t_1) = Wt \tag{28}$$

where H = heat added or subtracted from the water, Btu.
W = weight of water, pounds.
t_2 = highest temperature of water.
t_1 = lowest temperature of water.
$t = t_2 - t_1$ = temperature rise of the water.

The flow of heat from the engine to the cooling water is constantly taking place, and it becomes necessary to circulate the water through the engine-cooling system in order to prevent its boiling. Under this condition, therefore, Eq. (28) is considered to apply only for a unit of time, generally a period of 1 hr. Thus H would become the quantity of heat removed from the engine during a period of 1 hr and W would be the weight of water in pounds circulated during this 1-hr period. Since we are accustomed to thinking of water flow in terms of gallons per minute (gpm), the foregoing equation can be revised so that we can calculate directly the flow of cooling water in gallons per minute for a given heat dissipation per hour. Since a gallon of water at 60 F weighs 8.345 lb, and since a flow of 1 gpm is equivalent to 60 gal per hr, a rate of flow of 1 gpm is equivalent to

$$8.345 \times 60 = 500 \text{ lb per hr}$$

As previously given in Eq. (28)

$$H = Wt$$

Now since

$$W = 500\ (\text{gpm})$$

We can substitute in Eq. (28) and obtain

$$H = 500(t)\ (\text{gpm})$$

And by dividing we obtain

$$\frac{H}{00t} \qquad (29)$$

This equation can be solved graphically by the chart in Fig. 86. By means of a straightedge, connect the value for temperature rise desired and the heat dissipation to the cooling water per brake horsepower-hour and read on the scale for gallons per minute the quantity of water required to be circulated per brake horsepower. For a 10-deg. rise in the engine jackets with the cooling water required to absorb 3,000 Btu per hr per bhp, it will be necessary to circulate 0.6 gpm per bhp, and for a 1,000-hp engine, it becomes necessary to circulate 600 gpm of cooling water through the jackets. Any other combination of conditions can be determined rapidly from this chart.

All calculations for determining the quantity of water circulated through the jackets of an internal-combustion engine for cooling purposes involve the quantity of heat to be removed, the temperature rise permitted in the water being circulated through the engine, and the rate of water circulation. If the rate of water circulation is held constant, the temperature rise is directly proportional to the quantity of heat removed. The temperature at which the water enters the engine-cooling system has no influence on the temperature rise of the water in its passage through the engine. If the quantity of water circulated through the engine increases 10 F in temperature between the engine inlet and outlet at a given engine load, then that same 10 F differential will exist at that load regardless of whether the water reaches the engine at 100 or 175 F.

117. Cooling-water Temperatures.—The temperature at which water enters and leaves an engine should be given considerable study, much more in fact than has been accorded this important

subject in the past. Unfortunately, engine operators have been too prone to follow custom right or wrong, as a consequence the

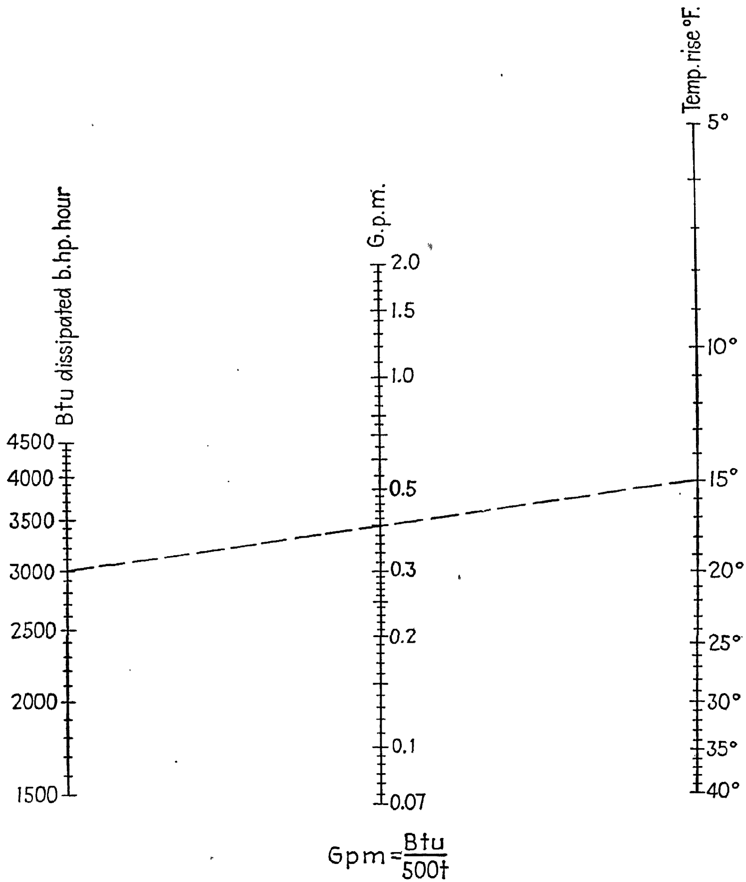


FIG. 86.—Nomographic chart for determining the quantity of water required for cooling engine cylinder jackets.

entire engine-cooling problem is still far from being on a rational basis. This lack of agreement is readily apparent from a study of the cooling-water temperatures employed in 145 oil-engine

plants reporting their operating results in the 1939 Report on Oil-engine Power Cost of the A.S.M.E., Fig. 87.

These data indicate that the temperature rise through engine jackets (difference between the temperature of the water into and out of the engine) varied from 5 to 90 F and that the temperature

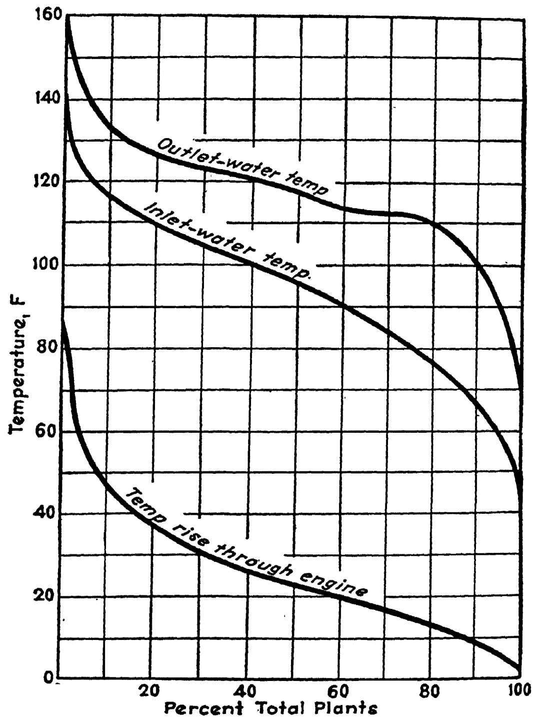


Fig. 87.—Variation in cooling-water-temperature conditions in diesel engines operating in central stations.

of the water leaving the engine varied from 70 to 160 F. One-half of the plants reporting gave temperature rises of less than 22 F. Over 85 per cent of the plants reported outlet water temperatures exceeding 100 F, while only 10 per cent reported outlet temperatures exceeding 125 F. These data represent the practice in plants using both open or single-circuit cooling systems as well as closed or double-circuit cooling systems.

When these data are divided between plants with single- and double-circuit cooling systems, there appears to be a distinct tendency for the higher outlet temperatures to be used only with the double-circuit cooling systems. In fact, 64 per cent of the stations reporting exit water temperatures over 120 F employed double-circuit cooling systems. While practically equal numbers of stations reported using single- and double-circuit cooling, the single-circuit systems were largely restricted to those plants

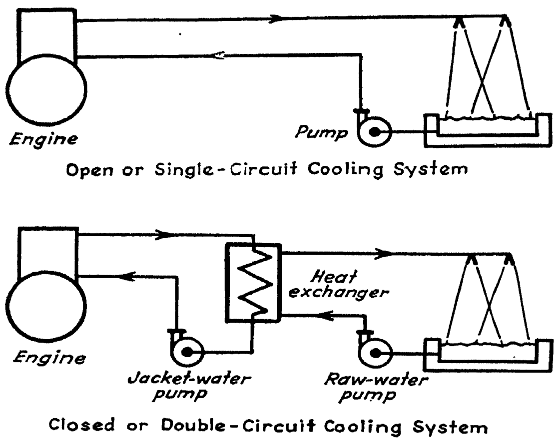


FIG. 88.—Schematic diagrams of cooling-water systems employed with internal-combustion engines.

where the outlet-water temperature from the engine was less than 120 F.

The foregoing analysis is fairly typical of the cooling conditions existing in the plants that the author has studied over the past decade. During those years when only single-circuit cooling systems were employed for engine cooling it was found advisable to keep the jacket-water temperature low in an effort to decrease the amount of scale deposits in the cooling-water passages of cylinder heads and cylinder walls. The advent of the double-circuit cooling system for stationary internal-combustion engines with softened water flowing through the engine jackets eliminated the excessive formation of scale and permitted the use of higher water

temperature to and from the engine jackets. Operators, schooled in the use of single-circuit cooling systems where the temperatures of cooling water had to be kept low, have been slow to take advantage of any possible economies from double-circuit cooling systems through the increase of jacket-water temperatures.

For many years it has been felt that the maximum temperature of the water leaving an engine should not exceed 120 F where a single-circuit cooling system was employed, and that where a double-circuit cooling system employing softened or scale-free water in the engine-jacket circuit was used it was safe to operate with a temperature of 140 F from the engine. Many operators, however, consistently maintain the temperature of the water leaving their engines at values considerably below these figures as is readily apparent from Fig. 87. *Temperatures of water to and from an engine which are maintained at too low a value may be as injurious to the engine as possible excessive temperatures.*

In an effort to clarify the somewhat scattered information now available on engine cooling-water temperatures, it will be necessary to consider the cooling limitations for several types of engines. Consideration will be given to the cooling-water conditions for

1. Slow-speed diesel engines.
2. High-speed stationary and automotive diesel engines.
3. Natural-gas engines.

It should be emphasized that the temperature rise of the water passing through the engine jackets bears no relation to the temperature at which water leaves the engine but is dependent upon the quantity of heat to be removed with a given rate of cooling-water flow through the engine jackets.

118. Cooling of Slow-speed Stationary Diesel Engines.—The cooling of large slow-speed engines involves consideration of the relative expansion of piston and cylinder liner as well as the effect of liner thickness on the cooling of the cylinder. Some engineers have used an empirical rule for determining the temperature of the water leaving engines of this type, as follows:

$$t_2 = 190 - 4D \tag{30}$$

where t_2 = temperature of water leaving engine.

D = cylinder diameter, inches.

In this equation it is assumed that the maximum possible outlet temperature from an engine is 190 F. This outlet temperature decreases uniformly with the increase in the cylinder diameter.

Many builders of large slow-speed engines for central-station service have required that the temperature of the water leaving the engine should not exceed 140 F, and in many instances, particularly with engines of 2,000 hp and larger, this outlet temperature has been held even lower. Apparently the reason for this request for low outlet-water temperatures is partially due to a desire to maintain reasonable lubricating-oil temperatures, and partially due to the relative expansion between cylinder liner and piston.

At the present time there is no agreement of opinion among designing engineers regarding outlet-water temperatures which are satisfactory for large internal-combustion engines. Perhaps this lack of agreement results from the fact that the internal-combustion engine, and particularly the large slow-speed units, are relatively new and the experience in cooling of such engines is not yet sufficient to justify a complete accord among designers in this important matter. During the past few years several engines in stationary service have been operating with outlet jacket-water temperatures of 175 to 180 F. While the experience with these slow-speed units of the diesel type has been highly satisfactory, nevertheless there is a hesitancy on the part of engine designers to increase outlet-water temperatures until such time as these trial installations prove themselves.

119. Cooling of High-speed Automotive-type Diesel Engines. The cooling of high-speed engines presents an entirely different problem from that involved in the cooling of large stationary engines. In the first place, space limitations on trucks and locomotives make it impossible to maintain cooling-water temperatures at the low levels which have been considered necessary with the large stationary engines. As a consequence it was necessary to design engines that would operate at higher cooling temperatures. In addition, as has been pointed out by Judge,[1] the temperature at which the cooling water is maintained in the engine jackets has a marked effect upon the power output and fuel consumption of the engine.

[1] JUDGE, ARTHUR W., "High Speed Diesel Engines," 3d ed., p. 39, D. Van Nostrand Company, Inc., New York, 1939.

As shown by Fig. 89, the horsepower output increased while the fuel consumption per horsepower decreased with an increase in temperature of the engine-jacket water. This action is contrary to the belief of many engine operators, but the experience as pointed out by Judge has been corroborated by the experience in the United States with high-temperature cooling of gas engines and high-speed internal-combustion engines burning oil. The matter of high-temperature cooling is becoming of such importance that a section is devoted to vapor-phase cooling where the jacket water is brought to boiling temperature.

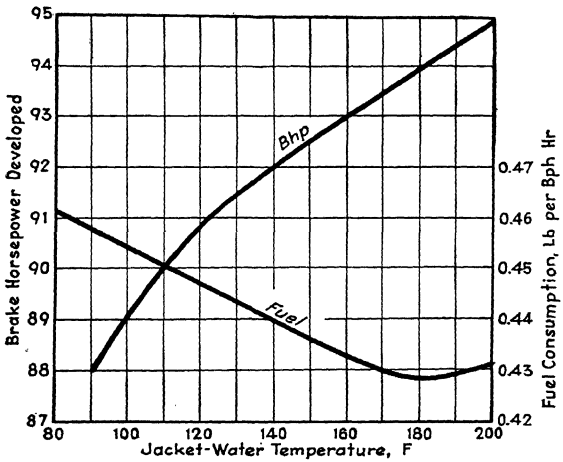


Fig. 89.—Effect of cooling-water temperature on power output and fuel consumption. (*Judge.*)

120. Vapor-phase Cooling.[1]—High-temperature cooling of small bore and stroke internal-combustion engines has been receiving considerable attention during the past two or three years, and if the commercial installations of this type of jacket cooling now in service prove successful, another milestone will have been reached in the improvement of internal-combustion engines. Essentially this cooling method consists of supplying water to the engine jacket at or near the boiling point. The

[1] Sanders, T. P., Vapor Phase Engine Cooling Introduced on Pipe Lines, *Oil Gas Jour.*, Sept. 19, 1940.

water is further heated by contact with the cylinder liner walls, and upon leaving the engine the pressure of the water is suddenly decreased, causing the water to flash into steam. The steam is in turn condensed in a radiator or heat exchanger and the condensate in the form of water returned to the cooling system to complete the cycle. By forcing the water through the engine jackets at a rapid rate and keeping the system, together with the circulating water pump, under a constant head, hot spots are avoided and a more uniform temperature is maintained over the entire length of the engine cylinder.

J. H. Wallace and R. A. Newton[1] have shown that the upper ends of the cylinder jackets in an internal-combustion engine are always at boiling temperature while in operation, regardless of the temperature of the circulating water. In other words, it is impossible to cool the tops of the cylinder liners with water and not have boiling take place on the outer surface of these liners. This same condition also applies to the water passages in the cylinder heads. Therefore, in order to reduce the temperature gradient between the upper and lower ends of the liners, it is necessary to increase the temperature at the lower ends rather than attempt to reduce the temperature at the head. By vapor-phase cooling the temperature differential over the length of the liners can be reduced more than 50 per cent, and since liner distortion is directly proportional to the temperature differential, the cylinder tapering due to temperature can be reduced more than 50 per cent by vapor-phase cooling.

Combustion of hydrocarbons in an engine cylinder produces carbon dioxide, carbon monoxide, water, and some small portion of sulfur dioxide. The quantity of water formed is surprisingly great, amounting to about 1 gal per 10 bhp-hr for gas engines and about ½ gal per 10 bhp-hr for diesel engines burning oil. Most of this passes through the exhaust valves as steam, but if the lower portion of the cylinder is below the dew point, some of the water vapor will condense there.

Moisture condensing on the cylinder wall quickly absorbs sulfur compounds to form sulfurous acid. When the rising piston rings scrape the condensate off the cylinder wall, it mingles

[1] *Oil Gas Jour.*, Sept. 7, 1939, p. 58. This same condition was also stressed by Wallace in a paper presented before the Tulsa, Okla., section of the Society of Automotive Engineers, Oct. 11, 1940.

with the oil film. The acid attacks walls, pistons, and rings and acts as an emulsifying agent to form sludge which accumulates in the crankcase and builds up a thick gummy deposit on rings and pistons.

The greatest benefit derived from vapor-phase cooling comes as the result of raising the wall temperatures in the lower portion of the cylinder to a point where condensation is avoided. In several modern installations, internal-combustion engines with this type of cooling are now operated on sewer gas or sour natural gas, both of which contain comparatively large portions of hydrogen sulfide. Even with fuels of this type, sludge formation is reduced to an almost inconsequential amount. An automobile equipped for vapor-phase cooling has been driven 25,000 miles without changing either the oil or the oil filter. The oil is still clear, save for a small accumulation of carbon particles.

Vapor-phase cooling can increase engine efficiency 3 or 4 per cent under full load or up to 8 or 10 per cent under partial load owing to the reduction in mechanical losses. The higher wall temperatures in the lower portion of the cylinder decrease oil drag and make for better lubrication because of the reduction in sludge formation. Reduction of the temperature gradient from top to bottom of the cylinder also serves to obviate some of the friction that results when piston rings are forced to expand and contact in a tapered cylinder.

In engines of the crank-pan type, such as automotive engines having underhung crankshafts, and most of the high-speed engines now used for stationary work, the oil contained in the crank pan is actually at a lower temperature when vapor-phase cooling is being used than when cooling water at a lower temperature is employed. Various explanations have been given for this apparent contradictory action, although the most logical explanation is that water at high temperature absorbs heat through the cylinder liners at a more rapid rate than it can do at lower temperatures and less heat finds its way to the lubricating oil.

In engines of the crankcase-type construction for stationary operation where the main casting is carried down to form the crankcase, there is usually an increase in the lubricating-oil temperature with vapor-phase cooling. This is due to the better conductivity of a large mass of cast iron and also to the fact that these engines generally do not have any air circulating about the

crankcase. Engines of this type are generally provided with lubricating-oil coolers. When engines of this type are converted to vapor-phase cooling, it is necessary to provide these lubricating-oil coolers with a separate source of low-temperature cooling water, or in the case of radiator-type engines a small section of the radiator is generally used for cooling the lubricating oil.

Experiments conducted in England show that with jacket-water temperature below 212 *F the liner wear in small cylinder sizes is*

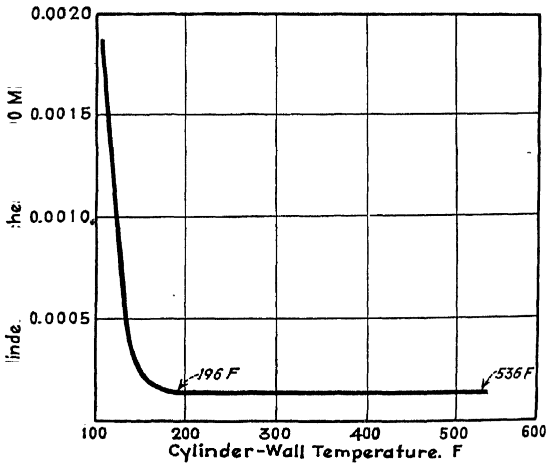
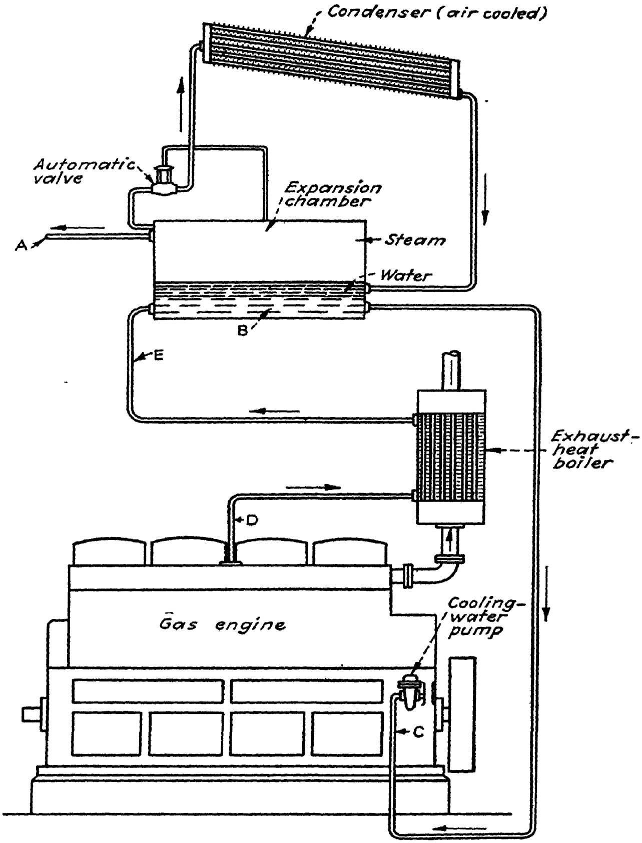


FIG. 90.—Cylinder wear in relation to wall temperature for automotive-type engines. (*Collected Researches on Cylinder Wear, Institution of Automobile Engineers, England.*)

almost inversely proportional to the lubricating oil consumed, while at temperatures above 212 *F the wear is independent of oil consumption, Fig.* 90.

Diesel engines have been operated in Great Britain with a steam pressure of 125 psi gauge on the jackets. This is a cooling temperature of 353 F. Liquid-cooled aircraft engines often operate at 325 F. Since there are no serious lubricating problems in either instance, these examples serve to indicate that cooling-water temperatures between 200 and 250 F are safe for all ordinary engines, provided that a sufficiently rapid rate of circulation is maintained.



TEMPERATURE AT VARIOUS POINTS
AT DIFFERENT STEAM PRESSURES

Steam pressure, lb g	Temperature F A	B	C	D	E
1	216	216	215	224	231
2	219	219	218	227	234
3	222	222	221	230	237
5	228	228	227	236	243

91.—Layout for vapor-phase cooling of internal-combustion engines. (*Courtesy of Power.*)

The American Gas Association[1] has made a thorough study of natural-gas engines using high-temperature cooling. The following summary is taken from that report:

Forty-six natural-gas engines, varying in size from 40 to 250 hp, have been observed over periods of from 12 to 18 months of operation at temperatures varying between 215 and 230 F. These engines have been periodically inspected during operation and have been torn down for detailed micrometer inspection for mechanical wear. Dynamometer tests have been conducted on engines equipped with conventional (low temperature or 160 F) cooling units, then rechecked after being cooled at temperatures above 212 F; the results have been as follows:

1. Dependability (continuity of service) has been as good, in all cases, and in many instances better, than that of engines operating at temperatures below 212 F.
2. Lubricating-oil consumption in engines operating with high-temperature cooling has been no greater than in engines cooled at lower temperatures.
3. Cylinder wear in engines cooled above 212 F has been normal, and in some instances, phenomenally low.
4. Piston rings in all engines operating above 212 F were entirely free and clean when inspected. (One exception was an engine operating in a dairy close to a beet-sugar pulp storage that had taken some of the fine pulp in through the crankcase breather, forming "molasses" in the oil.)
5. Engine-head gaskets of standard material have held perfectly in all cases.
6. Bearings inspected were in excellent condition, wear being normal or below normal.
7. Condensation (accumulation of water in oil sump) is entirely eliminated when engine-jacket-water temperatures are maintained about 212 F.
8. Dynamometer tests with and without high-temperature cooling show a slight increase in engine horsepower under full-load conditions with the same quantity of fuel, indicating possible reduction in "oil drag" at higher temperatures.

The foregoing results are of vital importance to the natural-gas power industry. It is now possible to safely advocate the use of high-temperature cooling wherever waste heat may be utilized to advantage in conjunction with the operation of a natural-gas engine.

[1] "Report of the Gas Engine Power Committee for 1940 and Census of Gas Engines," American Gas Association Committee Report.

121. Correct Cooling.—The proper cooling of an internal-combustion engine requires that the following conditions be met:

1. There must be a continuous uninterrupted flow of water or other suitable cooling medium through the engine jackets during the time the engine is in operation and, in most cases, for some time after the engine stops operating.
2. The temperature rise of the water passing through the engine should be limited to not more than 20 F except in rare cases.
3. The water used for cooling in the engine jackets should be reasonably free from scale-forming impurities. The impurities should never be allowed to become of such quantity that scale deposits sufficient to interfere with proper cooling are created in the water passages.
4. The water used in the engine jackets should not be corrosive to the metals with which it comes in contact, through either oxygen corrosion or other forms of corrosive action.
5. The temperature of the water leaving the engine should not exceed the limits imposed by the manufacturer of the engine.
6. Provisions should be made for maintaining a constant water temperature either to the engine or from the engine through the use of suitable thermostat control. It is impossible to maintain both outlet and inlet water temperatures constant unless the engine is pulling a constant load. The difference between the inlet and outlet temperatures (temperature rise) will vary with the load on the engine; the greater the load, the greater the temperature rise of the water passing through the engine.

122. Cooling Methods.—Cooling methods used in any particular case will be influenced by the character, quantity, and type of cooling mediums available. Thus cooling ponds, cooling towers, rivers, lakes, oil flowing through a pipe line, radiators, or other cooling sources have been used. For example, a plant located on a river having sufficient flow could utilize river water as a cooling medium in a closed cooler, while the engine in a pipe-line pumping station would utilize the oil or gasoline being pumped through the line for cooling purposes.

Few plants are so favorably situated that they can use a river or lake for a cooling source. Consequently it is necessary in

most instances to provide special cooling facilities such as a spray pond, cooling tower, or cooling pond.

Where an internal-combustion-engine power plant is operated by a municipality in conjunction with a water-works system, it is sometimes possible to use the raw water pumped to the water-treating plant as a source of cooling for the internal-combustion engines.

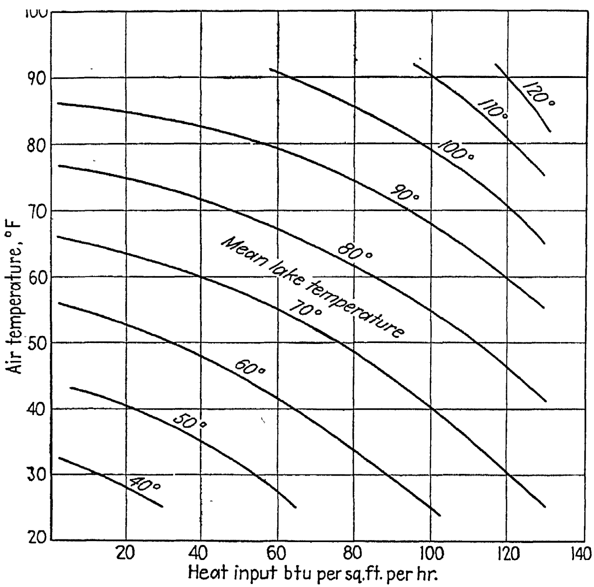


FIG. 92.—Mean lake temperature vs. heat input and air temperature in cooling ponds, wind velocity 6 mph. (*Courtesy of Power.*)

123. Surface Cooling Ponds.—Surface cooling ponds do not involve the use of any mechanical equipment aside from a suction and discharge well. Having only its surface exposed to the air, the cooling efficiency is extremely low, Fig. 92, necessitating a very large area if water temperatures are to be held to desirable levels. This method of cooling is generally not considered reliable because of the possibilities of sudden water loss, earthing in of basin, and the great amount of heat absorbed from the sun. Algae growth, icing, and building up of concentrates

are other features difficult or costly to control. Maintenance costs vary too widely with a given installation to be comparable. Low pumping head and absence of drift losses are desirable features.

124. Spray Ponds.—A spray pond consists essentially of a group of spray nozzles supported above a pond or basin, and the cooling of the water is effected through evaporation of a portion of the water and through the cooling effect of wind playing over the pond surface. There is a wide difference in cooling perform-

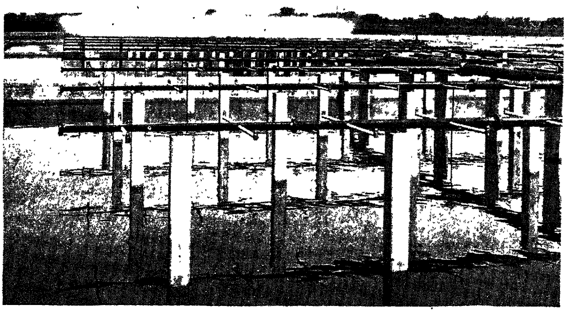


FIG. 93.—Large spray pond with part of the pond removed from service. (*Courtesy of The Marley Company.*)

ance between the very small and the extremely large pond. The smaller spray pond is capable of somewhat better cooling and has a lower first cost, although it will show a greater drift loss when compared with a large pond on a gallon-per-minute–area basis. In comparing spray ponds with other types of cooling the characteristics of a pond of average size are usually considered. Spray ponds of practical design are limited in cooling efficiency, which falls off rapidly with a decrease in wind velocity or adverse wind direction or both. Drift loss as well as ground area required are comparatively large for the ordinary spray pond, although drift loss can be minimized by the use of an effective louver fence around the pond which adds materially to the cost

of the installation. First cost and maintenance costs are greatly affected by the basin design. The pumping head is relatively low resulting in economical pumping costs. Although spray ponds were widely used and still have a definite and important place in industrial water cooling, the number of installations being made today has sharply decreased since cooling towers have been demonstrated to have superior efficiency and economies under most operating conditions.

125. Spray Towers.—Spray towers are in effect small spray ponds closely surrounded by a louver fence. The spray system, however, is located at the top of the tower and is directed downward in most cases. The efficiency of spray towers is better than that of spray ponds since they are not sensitive to wind velocity and direction. This is due to the induction of air by the pressure of the falling spray water in the tower during periods of wind stagnation or adverse direction. Drift loss is quite small from towers with effective louvered sides, and such fine moisture as does escape is usually confined to the immediate site.

Reasonably low pumping head, simplicity of design, ease of operation, and slight maintenance are desirable features. The original cost for the small spray tower is exceptionally low but increases with size to the point where a unit having a capacity of 1,600 gpm or more is not always economical. Large-sized towers are very long owing chiefly to the fact that in order to make efficient use of the breeze through the tower, it must be kept narrow.

126. Spray-coil Towers.—Spray-coil towers are spray towers with atmospheric heat-exchanger coils in the base. The spray tower cools the raw water which in turn cools the heat-exchanger coils as it is showered over the exterior of the tubes. Soft water, oils, and other liquids as well as gases are cooled by being circulated through the coil tubes. This combination of spray tower and atmospheric heat exchanger is particularly advantageous in that the medium being cooled within the exchanger is neither contaminated by contact with air nor lost by evaporation or drift. When water is the medium being cooled inside the exchanger tubes, the building up of concentrates, sludge, and scale-forming substances is eliminated.

Spray-coil towers are widely used in the indirect cooling of jacket water for diesel and gas engines, compressors, and similar

applications. Atmospheric heat exchangers are also combined with deck and mechanical-draft towers where the application demands that the tower have certain features not applicable in the spray type.

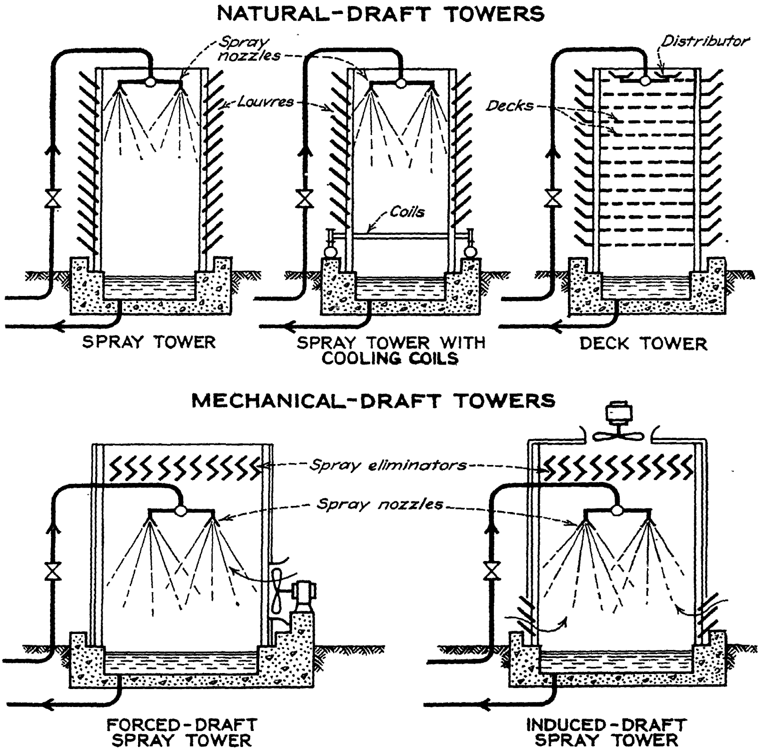


FIG. 94.—Cross sections of typical natural- and mechanical-draft towers. (*Courtesy of Power.*)

127. Deck Towers.—Deck towers depend for their cooling effect upon the falling of the water through a series of staggered trays or wood deflecting plates. They can be operated at high cooling efficiency provided that wind of appreciable velocity and from the proper direction is present, since the cooling efficiency of this type of equipment drops rapidly as wind velocity falls

below that upon which the tower performance is based. This sensitivity to wind velocity is due to the necessity of having air movement across the horizontal decks. Air movement by induction or convection is not rapid enough to be effective during periods of calm.

Drift loss is small in comparison to that experienced with the spray pond and is less than for spray towers. This loss increases as wind velocity increases but not always in direct

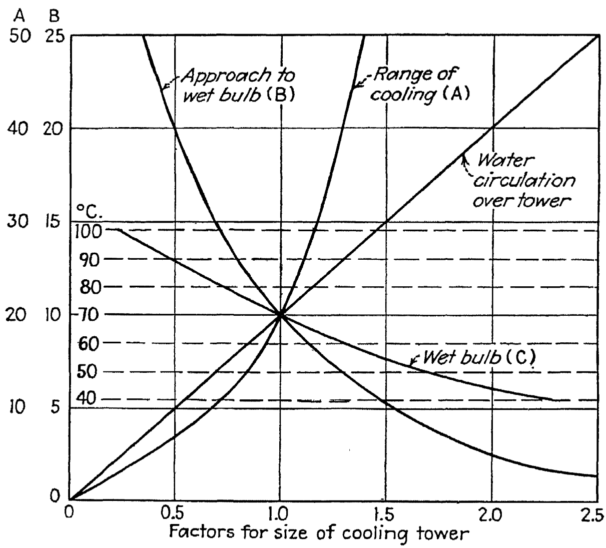


FIG. 95.—Variables affecting size and operation of cooling towers. (*Bischof.*)

proportion. The design of the louvered sides of a deck tower is a compromise between providing the freest passage of air and holding to a minimum the amount of water that escapes during high wind velocities. The required pumping head varies with the tower height and is, on the average, greater than for other types of cooling equipment. Like spray towers, the deck type must be built long and narrow. First cost gradually decreases from a relatively high figure for smaller sizes to a point where the price of a tower having a capacity of approximately 1,000 gpm or more is quite moderate.

128. Mechanical-draft Towers.—Mechanical-draft towers, either forced- or induced-draft type, employ mechanically driven fans to create the air currents required for cooling. They can be designed for high cooling efficiency under all conditions since the efficiency is entirely independent of prevailing wind velocities or direction and is controlled or varied by the speed of the fans as well as the number of fans in operation. Drift loss is almost negligible. Ground-area requirements are small in comparison with ponds or deck towers.

Pumping-head requirements are somewhat high, but not so great as for deck-type towers. First cost, based upon comparative performance during all wind conditions, is low and particularly so for the larger installations. The additional power consumed for fan operation is usually offset by the savings effected because of the higher efficiency obtained.

This type of equipment has been measurably improved during the past few years. It largely avoids the objectionable characteristics of other types of equipment such as drift loss, large ground area, poor appearance, and unreliable performance during low wind velocities. The large number of mechanical-draft-tower installations being made is evidence of their surpassing features and qualities.

It is advisable to have a minimum of two fans installed on a mechanical-draft tower. The provision of at least two fans is an insurance against the possibility of extremely poor cooling in the event of fan or fan-motor failure. In a tower with a single fan, the inability of the fan to operate reduces the cooling capacity of the tower materially. With at least two fans installed in the tower, the failure of a single fan during periods of extreme cooling requirements would not decrease the cooling capacity of the tower to the extent that would occur if only one fan were installed.

129. Evaporative Coolers.—Evaporative coolers have been developed to meet the need for a compact and efficient water cooler in the internal-combustion-engine field. It is in effect a double-circuit cooling system and takes the place of the cooling tower and heat-exchanger combination. Softened jacket water is pumped through the cooler coil of either fin-tube or straight-tube design constructed of copper or other suitable metal. The coil is kept moistened from a spray system supplied by a small centrifugal pump which recirculates water from the supply

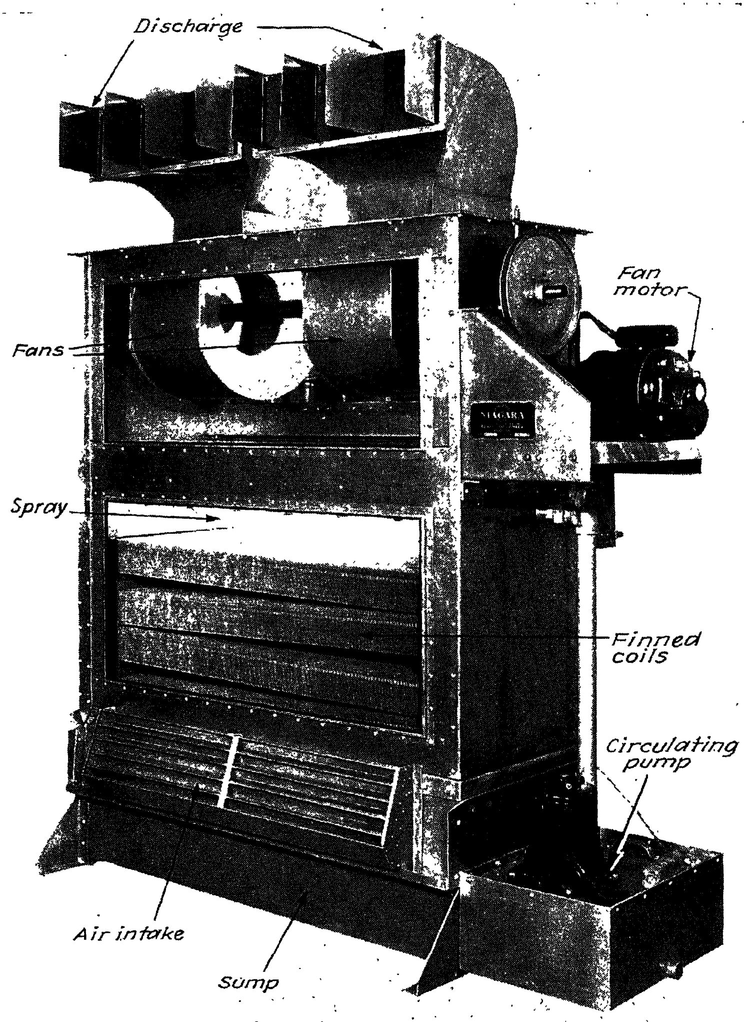


FIG. 96.—Demonstration model of evaporative cooler showing construction details. (*Courtesy of Niagara Blower Company.*)

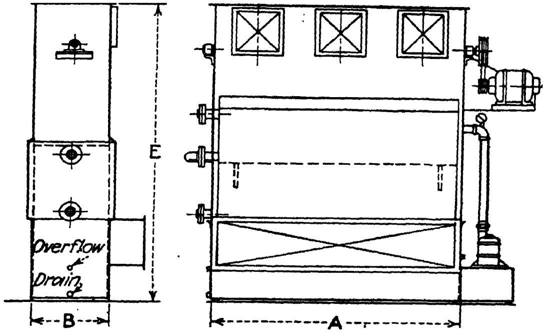


Fig. 97.—Performance and dimensions of evaporative coolers. (*Courtesy of Niagara Blower Company.*)

Dimensions in inches—side air discharge models with inlet duct connections			
Model no.	*A*	*B*	*E*
20	51½	20½	88
25	70½	20½	88
30	70½	30½	102½
35	70½	30½	102½
40	94½	30½	102½
45	94½	30½	102½
60	109½	45½	132¼
75	109½	45½	132¼

Capacities When Used for Water Cooling

Model number	Gpm water circulated												
	20	40	60	80	100	120	140	160	200	250	300	350	400
20	2630	3700	4250										
25	....	4050	4700	5100	5400								
30	....	4600	5500	6130	6550	6840							
35	....	5150	6300	7000	7500	8000	8300						
40	....	....	6700	7500	8100	8600	8950	9270					
45	....	....	7500	8500	9220	9800	10300	10750	11500				
60	....	....	8250	9660	10700	11400	12000	12500	13300	14100	14700	15200	15500
75	....	....	8850	10700	12150	13250	14050	14740	15950	17300	18500	19400	20100

Factor Table[a]

Entering air wet bulb, deg. F	Water temperature entering condenser, deg. F						
	100	110	120	130	140	150	160
60	87	113	138	163	185	206	223
65	77	104	130	154	177	198	216
70	68	95	120	145	168	189	208
75	60	86	112	136	160	181	200
80	50	77	103	127	151	172	190

[a] To obtain Btu per hour capacity of liquid cooler, multiply factor corresponding to ntering water temperature and entering air wet bulb temperature by the basic rating of the liquid cooler at the quantity of water given.

Water Friction Loss—Table in Feet Head

Model number	Gpm water circulated												
	20	40	60	80	100	120	140	160	200	250	300	350	400
20	1.0	4.0	9.2	16.3									
25	...	4.8	11.2	19.6	29.6								
30	...	2.8	4.8	8.8	13.6	19.2	25.6						
35	...	2.8	4.8	8.8	13.6	19.2	25.6						
40	...	...	5.6	9.6	14.8	21.2	28.8	37.2					
45	...	...	5.6	9.6	14.8	21.2	28.8	37.2	12				
60	...	...	4.4	7.2	10.4	14.8	19.2	25.2	38.4	8	12	16	21
75	...	...	4.4	7.2	10.4	14.8	19.2	25.2	38.4	8	12	16	21

tank in the bottom of the cooler. Air is drawn over the wetted coil by means of a pressure-type fan, and evaporation of water on the coil surface produces the cooling of the jacket water inside the coil.

Water to replace that lost from the supply tank through evaporation on the surface of the coil is furnished by means of a supply line controlled through the operation of a float valve. This make-up water should be softened to prevent the formation of scale on the fin tubes through evaporation. The amount of make-up water required is approximately 1 gpm for each 500,000 Btu dissipated per hour.

Essentially this type of unit employs a relatively small quantity of water on the cooling side together with a large volume of air. For example, a unit with a cooling capacity of 3,000,000 Btu per hr for a 1,000-hp engine operating at full load would evaporate water at the rate of 6 gpm and circulate air at the rate of 14,000 cfm in the process.

In locating a unit of this character, which is usually designed to be installed inside the power plant or engine room, provisions should be made to take the required circulating air both from the outside and inside of the power plant. During the summer months a unit of this type can be used for ventilation by taking air from inside the engine room, and in the winter a major portion of the air requirements can be taken from outside the building. It is necessary to temper the air taken from the outside when it drops to 35 F by the inclusion of air from inside the building to prevent freezing of the spray system. By shutting off the water sprays and circulating the air in the building through the cooler, it can serve to heat the building during winter periods.

130. Radiators.—Radiators of the fin and tube type, comparable in general appearance to the radiator on an automobile, have been used for internal-combustion-engine cooling in both stationary and mobile service. Units of this type depend entirely upon cooling the water in the coils by raising the temperature of the air passing over them. Since the specific heat of air is relatively low, requiring only 0.01812 Btu to raise a cubic foot of air one degree at 70 F, the quantity of air that must be circulated becomes very great when any extensive amount of cooling is required. For example, if the air temperature is 70 F with the average water temperature in the cooling radiator at

120 F, and with 3,000 Btu removed per engine horsepower, it would require the circulation of approximately 120 cfm of air to remove this quantity of heat. Thus to cool the water for a total of 1,000 bhp operating would require the circulation of 120,000 cfm of air through the radiator.

It is readily apparent that radiator cooling is best adapted to small-engine installations. When engines are large (500 bhp and larger), and particularly when several large engines are operating together, the necessary radiator capacity for cooling becomes too unwieldy for satisfactory operation in comparison with other satisfactory cooling methods available. For the small installation, however, the radiator can be used in many instances very advantageously.

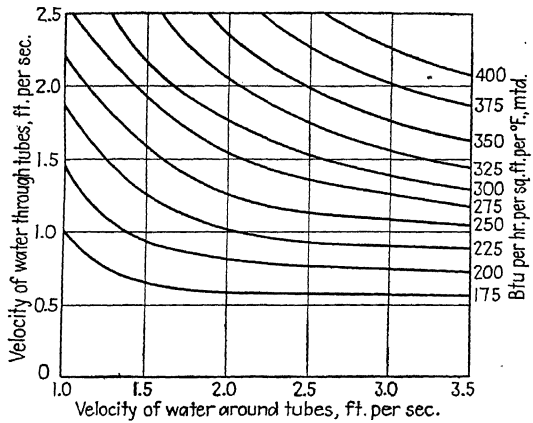


FIG. 98.—Heat-transfer rate for shell-and-tube-type water coolers. (*Courtesy of Schutte & Koerting Company.*)

131. Closed Tubular Heat Exchangers.—Shell and tube heat exchangers are used to advantage for cooling the engine-jacket water in water-works and pipe-line installations. In these instances the heat-absorbing medium is the water being pumped for human consumption in the case of the water-works installation, or the petroleum product being pumped through the pipe line in case of the pipe-line pumping station.

The cooling area required in a shell and tube heat exchanger can be approximated by the use of data for heat absorption

contained in Fig. 98. The mean temperature difference can be obtained from the equation

$$\text{MTD} = \frac{\theta_a - \theta_b}{2.3 \log \left(\dfrac{\theta_a}{\theta_b}\right)} \tag{31}$$

where θ_a = greatest temperature difference.
θ_b = least temperature difference.

If, for example, jacket water from the engine enters the cooler at 145 F and leaves at 135 F and the raw water enters at 80 F and leaves at 95 F, then

$$\theta_a = 145 - 80 = 65$$
$$\theta_b = 135 - 95 = 40$$

and

$$\text{MTD} = \frac{65 - 40}{2.3 \log (65/40)} = 52$$

For a water velocity of 2 fps through the tubes with a velocity of 1.5 fps for the raw water around the tubes the heat transfer would be 310 Btu per sq ft per hr per deg MTD (Fig. 98) and the total heat transfer under the conditions considered would be 52 × 310, or 16,120 Btu per sq ft per hr. Fouling of the heat exchanger in service must be allowed for, and it would be safe to assume that the heat transfer would be 75 per cent of the value computed under average conditions.

132. Cooling-tower Basins.—Basins for either spray ponds or cooling towers deserve considerable study, both from the standpoint of the hydraulics involved in the circulation of water through the cooling-system piping in the plant and from the standpoint of basin cleaning while maintaining the cooling system in service.

An examination of many basins for both spray ponds and cooling towers indicates that a gradual accumulation of dirt and rubbish in the basin occurs regardless of the care that is exercised by the plant operating staff in the maintenance of the cooling system. This extraneous material has the unfortunate habit of accumulating on or near the intake to the pump suction of the cooling system and, in several instances, has effectively stopped water flow from the basin. Furthermore, in those cases where a

basin for the cooling system is installed in a locality where dirt can be blown into it, considerable difficulty is experienced with clogging of pumps, valves, fittings, and strainers.

These conditions, which have been observed in operating practice, have caused the organization with which the author is associated to provide divided basins for cooling towers and spray ponds arranged so that one-half of the basin can be taken

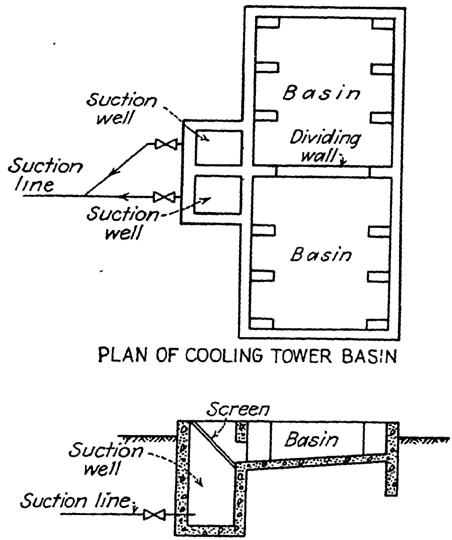


FIG. 99.—Details of divided cooling-tower basin so arranged that one-half of the basin can be taken out of service for cleaning.

out of service, cleaned, and put back into operation during low-load periods when the demand for cooling is not equivalent to the maximum capacity of the tower or pond. By taking one-half of the basin at a time, it is possible to clean it effectively, keep cooling service up to the requirements of the station at all times, and eliminate the danger of a cooling-water shortage due to the stoppage of a pump or plugging up of a valve, line, or strainer. The ability to remove one-half of the basin from service for cleaning, thereby discarding the water contained in that half,

has the further advantage of reducing the concentration of mineral salts in the water when the basin is replenished after cleaning.

A typical construction for such a cooling-tower basin showing the division of the basin as well as the pump suction well is shown in Fig. 99. In this particular construction, the basin floor slopes toward the suction well in order that dirt and rubbish will accumulate in the low portions where it can be more readily cleaned out. Each half of the basin is equipped with an intake screen arranged so that the entire suction well is protected from falling leaves and other debris. The sloping screens have the further advantage of being more easily cleaned than a vertically set screen.

133. Cooling by Evaporation.—The temperature to which water can be cooled by evaporation in either a spray pond or cooling tower is limited by the wet-bulb temperature prevailing in the locality. Wet-bulb temperature[1] is that temperature attained by a thermometer with its bulb kept constantly moistened as differentiated from normal atmospheric temperatures determined with a thermometer having a dry bulb.

While the wet-bulb temperature is the lowest limit to which water can be cooled by evaporation, this temperature is only approached but never realized in practice. Economic considerations in the application of either spray pond or cooling tower limit the closeness with which the temperature of the cooled water approaches the wet-bulb temperature. It is economically possible with a mechanical (either forced or induced) draft tower to cool water to within 10 F of the wet-bulb temperature, while spray or deck-type towers can cool to within 15 F of the wet-bulb temperature economically. Water in a spray pond can be cooled only to within 16 to 18 F of the wet-bulb temperature.

A valuable aid in the determination of the cooling limitations of any evaporative cooling equipment, such as spray ponds or cooling towers, are the summer wet-bulb temperature data for the United States given in Fig. 100. For example, this map shows that Evansville, Ind., has a maximum wet-bulb temperature

[1] MILLIKAN, ROBERT ANDREWS, HENRY GORDON GALE, and CHARLES WILLIAM EDWARDS, "A First Course in Physics for Colleges," p. 282, Ginn and Company, Boston, 1928.

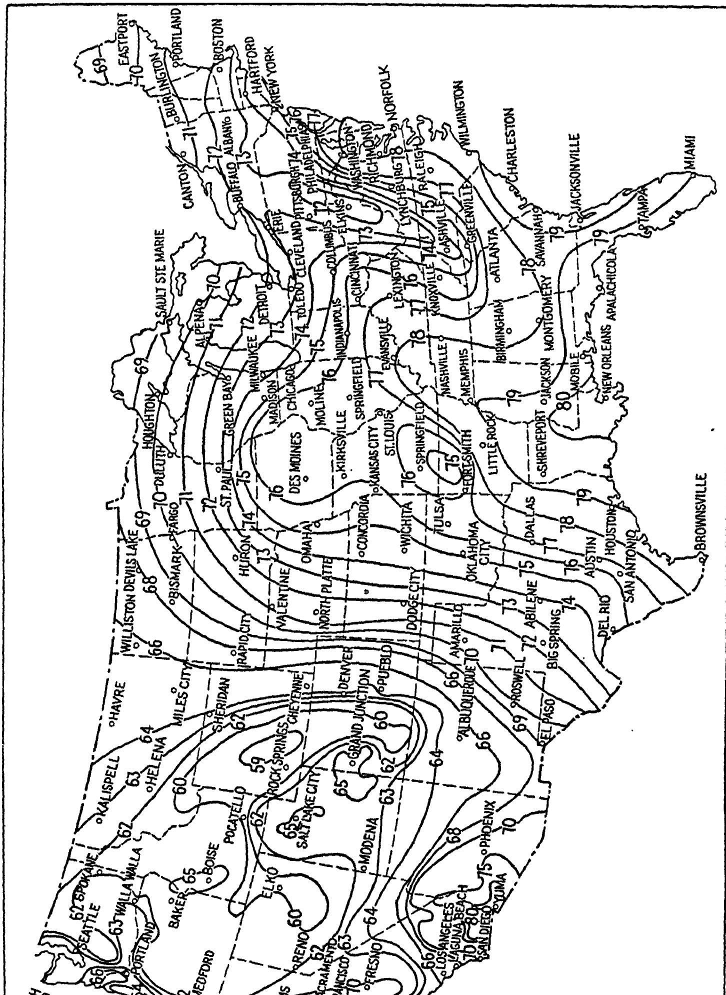


Fig. 100.—Summer wet-bulb temperatures. (*Courtesy of The Marley Company.*)

of 78 F, while Denver, Colo., has a maximum of 64 F. Thus with a mechanical-draft tower, water can be cooled to 78 + 10, or 88 F, at Evansville, while at Denver the temperature of the water can be reduced to 64 + 10, or 74 F, during the summer months.

The values for wet-bulb temperatures shown on the map will not be exceeded more than 5 per cent of the time during the months of June to September. An analysis of experience with evaporative cooling installations in various parts of the United States indicates that designs based upon wet-bulb temperatures which are not exceeded more than 5 per cent of the time during these months give satisfactory results without requiring an excessive investment in cooling facilities.

134. Proportioning Pipe Sizes.—What pipe size should be used for cooling-water lines? The size line employed in any cooling system should be such that the required quantity of water is circulated for the lowest cost. If the pipe size employed is too small, an excessive amount of power will be required for pumping. Should the pipe size be too large, the fixed charges on the piping investment will be larger than any possible savings in power costs. The proportioning of the piping for the engine-cooling system, therefore, resolves itself into consideration of both economics and engineering design.

Power required for circulating water through any piping system is employed to lift the water from its initial position to the position desired and to overcome the friction resisting the flow of the water through the pipe, fittings, and valves forming the transportation system. The difference in elevation between the point from which the water starts and the point where it is desired is known as *static head* and is usually expressed in feet of water. The friction loss in transporting the water through the pipe is known as *friction head* and is also usually expressed in feet of water.

If water is pumped through a horizontal pipe and discharged at the same elevation as it entered the pump suction, there is no static head to overcome, and the power required for pumping is used to overcome the friction head of the piping system. A similar condition may occur in the jacket-water side of a closed cooling system where the pumps take suction from a hot well and after passing the water through a cooler and then through

the engine discharge it back into the hot well again. The pumping of the water over a cooling tower represents a condition where both static head or the height of the point of water discharge above the water level in the tower basin and friction head or the friction in the piping system are present.

135. Piping Losses.—Determination of losses in the piping system can best be shown by means of an example.

Consider the layout of a single-circuit or open cooling system for an engine about which the following information is known:

Engine brake horsepower	500
Heat dissipation per brake horsepower-hour	3000 Btu
Temperature rise of cooling water	15 F
Water circulated per brake horsepower (see Fig. 86)	0.4 gpm
Total water circulated (500 × 0.4)	= 200 gpm

Consider also that the following quantity of pipe and fittings are required to form the system.

Pipe, linear feet	300
90-deg elbows	8
Gate valves	2
Swing check valves	1
Entrance from basin	1

It is also known that the static head over the cooling tower in the system is 30 ft and that the loss through the engine jackets is 7 ft.

Friction loss in fittings can be determined in several ways, although one of the simplest is to express the resistance in terms of equivalent length of straight pipe. The nomographic chart, Fig. 101, gives the resistance of various types and sizes of fittings and is satisfactory for the hydraulic calculations involved in the layout of internal-combustion-engine cooling systems.

In the example under consideration, calculations of the head loss through pipes of 3, 4, and 6 in. will be made for comparative purposes. All calculations can be shown in Table 34.

The first step is to determine the equivalent length of the piping system in terms of feet of straight pipe. In this calculation, the resistances of the valves and fittings are converted to equivalent lengths of pipe by the use of the chart, Fig. 101, and these equivalent lengths added to the length of pipe in the system to obtain the total equivalent length. Thus the equivalent length of 3-in. pipe is 376.3 ft; 4-in. pipe, 410.5 ft; and 6-in. pipe, 471 ft.

The second step is to determine the friction loss when passing 200 gpm through the piping system. Friction losses in pipe ranging from 1 to 42 in. diameter and for rates of flow from 1 to 100,000 gpm are given in Fig. 102. This chart is based upon the Hazen-Williams equation for friction losses in

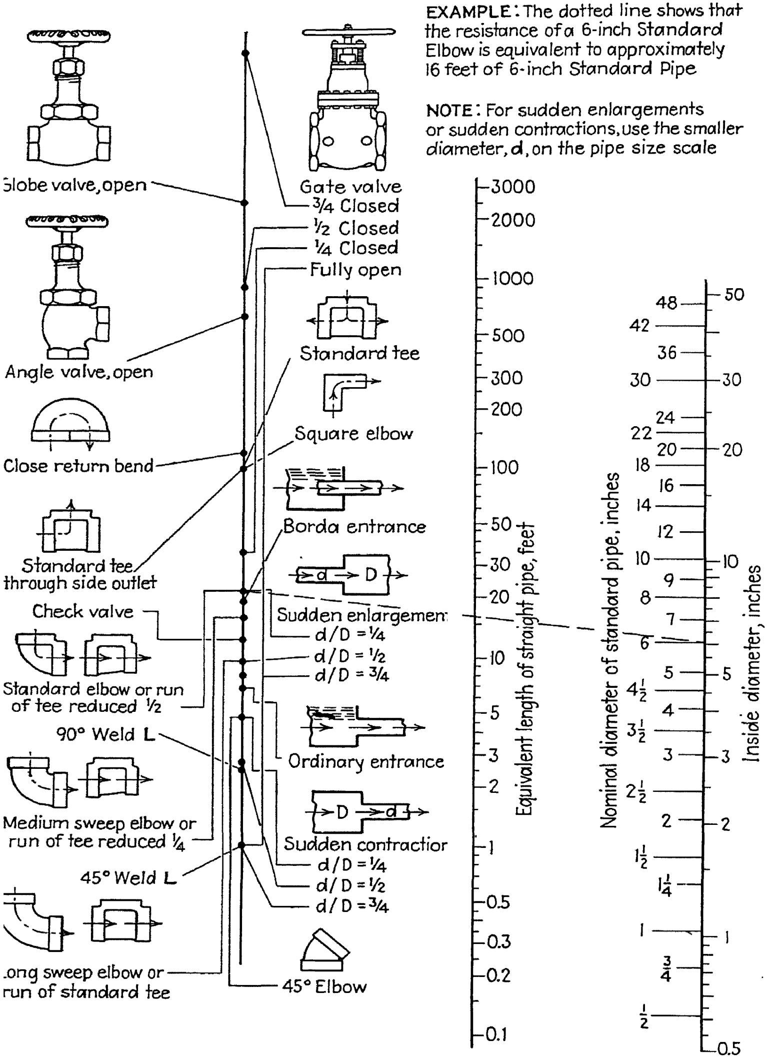


FIG. 101.—Resistance of valves and fittings to flow of fluids. (*Courtesy of Crane Company.*)

pipes. Starting from the value of rate of flow in gallons per minute at the bottom of the chart, proceed vertically upward until the diagonal line for the pipe diameter is reached. From this intersection point proceed horizontally either right or left and read the head loss expressed in feet per 100 ft of pipe. Thus the friction loss when passing 200 gpm through a 3-in. pipe is 18 ft per 100 ft of pipe, and the friction loss for the other pipe sizes as taken from the chart are shown in Table 34. The total pipe friction is obtained by multiplying the equivalent pipe length, expressed in hundreds of feet, by the friction loss per 100 ft. To this is added the friction loss through the engine and the static head of the cooling tower to obtain the total pumping head.

TABLE 34.—CALCULATION OF HEAD LOSS IN PIPES

Pipe diameter, inches	3	4	
Flow, gpm	200	200	200
Velocity, feet per second	9.08	5.11	2.27
Length of pipe, feet	300	300	300
Equivalent pipe length for:			
Eight 90-deg elbows	60		136
Two gate valves	5.6	8	13
One check valve	6	8	12
One entrance	4.7	6.5	10
Total equivalent length of pipe	376.3	410.5	471
Head loss per 100 ft, feet (Fig. 102)	18	4.4	0.61
Pipe friction loss, feet	68	18.1	2.9
Friction loss through engine	7	7	7
Static head over tower	30	30	30
Total pumping head, feet	105	55.1	39.9
Relative pumping head, per cent	100	52.5	38.0

From this series of calculations it is seen that increasing the pipe diameter 3 to 4 in. practically cut the pumping head in half, while increasing 4 to 6 in. only reduced the total pumping head about 28 per cent.

136. Pumps.—Cooling-water pumps used in internal-combustion-engine plants are practically all of the centrifugal type since this type is simple in construction, low in cost, and can operate under a wide variety of conditions. Liquid entering the center of the rotating impeller is accelerated to a high velocity and discharged by centrifugal force into the pump casing and out the discharge. For most conditions encountered in internal-combustion-engine plants, the single-stage single-suction pump will suffice since its total discharge head is usually sufficient and the quantity of water handled is not so great that the double-suction type is required.

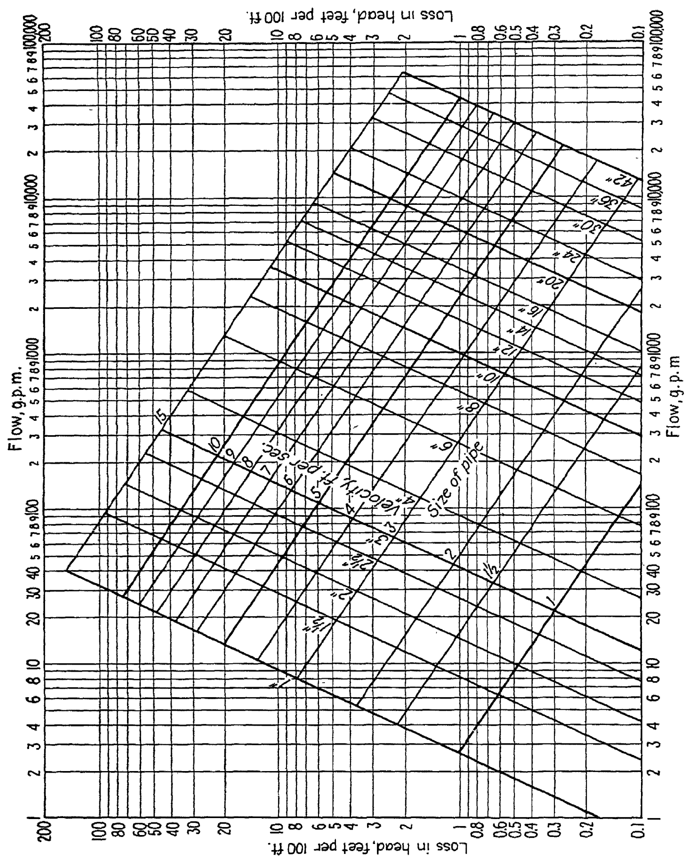


Fig. 102.—Chart for loss of head for water flowing through pipes. Based upon Hazen-Williams formula.

Constant c	Correcting factor	Type of pipe
140	0.54	Very smooth and straight WI or brass pipe. Very best and new straight CI pipe
130	0.62	Ordinary straight brass pipe or copper tubing or good new CI pipe
120	0.71	Smooth new WI pipe or 4- to 6-yr-old CI
110	0.84	Spiral riveted steel pipe; flow with lap or 10- to 12-yr-old CI pipe
100	1.00	Ordinary WI pipe spiral riveted steel pipe flow against lap. 13- to 20-yr-old CI pipe
90	1.22	26- to 30-yr old-CI pipe

Note: Correction factor applies to head loss only.

A trend in the construction of the single-stage single-suction pump is to provide the suction in the center of one side of the pump casing with the discharge leading from the pump casing in whichever direction around the periphery of the pump is found convenient. This type, known as the single-stage end-suction pump, is usually mounted directly on the motor driving it, permitting installation in a minimum of space and eliminating checking of alignment of pump and motor during installation. Where a centrifugal pump employs side suction and side discharge, it is found advisable to equip the pump with a split casing so that the upper half can be removed for inspection and maintenance as the occasion may require.

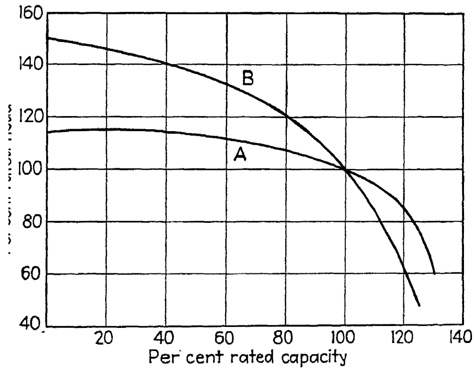


FIG. 103.—Probable range of head-capacity relationship for centrifugal pumps handling water.

Since the centrifugal pump moves the liquid by imparting a velocity to it and not by direct displacement as is the case in a plunger pump, the quantity of liquid which a particular centrifugal pump will discharge in a given time is influenced by the pressure or head against which the pump must operate. This relationship between the rate of liquid discharge and the head against which it is discharged is determined by the design of the particular pump and can be varied in the design within certain limits. The approximate range in the relationship between the pumping head and capacity of the pump is shown in Fig. 103. Curve A is the head-capacity curve of a pump having a flat-head

characteristic, *i.e.*, a considerable variation in the rate of discharge of water occurs with a slight change in the pump discharge head. An increase of 5 per cent in the discharge head will decrease the quantity of fluid pumped by 15 per cent, while an increase of 15 per cent in head will reduce the pump discharge rate 70 to 80 per cent. It will be noticed further that the discharge head is practically constant for rates of discharge ranging from no flow to 40 per cent of rated pump capacity. On the other hand, curve *B*, for a pump having a steep-head characteristic, indicates that an increase of 15 per cent in the pumping head will only reduce the discharge rate about 13 per cent. In order to stop the pump from discharging water, it is necessary to increase the head 50 per cent above that for which the pump was designed. The discharge head for zero flow is known as the *shut-off* head.

In many instances, pumps having a characteristic similar to that shown in curve *A* have been installed for operation in parallel. After installation it was discovered that practically no more water could be discharged from the two pumps operating in parallel than could be discharged from a single pump. The reason for this condition lies in the fact that when the rate of water flow through the piping system increased the friction head on the system increased. With an increase in the head against which the pumps must operate, the rate of pumping decreased. This increase in friction head, and consequent increase in total pumping head, has generally dictated the use of steep-head characteristic pumps for cooling-water service. It is only after a thorough analysis of the entire hydraulic system, including friction and static head losses of the piping as well as the head-capacity characteristics for several pumps, that the correct head-capacity characteristics for the pumps can be determined.

The horsepower required to drive a pump is determined by means of Eq. (1), Chap. V.

137. Selection of Pumps.—As previously pointed out, the selection of the correct pump requires a study of the hydraulic characteristics of the system and the head-capacity characteristics of the pumps used to circulate the water through the system. The first step necessary is to determine the head-capacity characteristics of the system through which the water is being pumped. In Fig. 104 is shown the head loss through

the raw-water side of a cooling tower used for a closed or two-circuit cooling system in a typical case. The curve for the total head loss through the system shows that with no water flowing the head on the system is 27 ft. This is the static head to which must be added the friction head of the piping system for different rates of flow through it. Thus at a flow rate of 600 gpm the total head is 29 ft, while at 1,000 gpm it is 33 ft, and at 1,500 gpm it is 40.5 ft. The problem becomes one of selecting suitable pumps to deliver quantities of water varying from 500 to 1,500 gpm as required by the plant operating conditions.

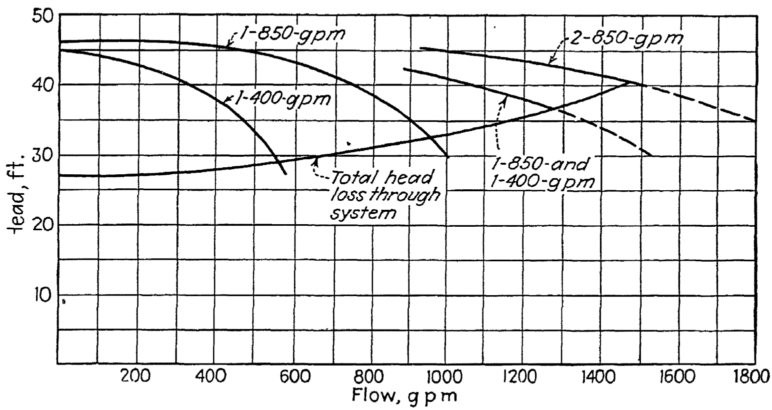


Fig. 104.—Head-capacity curves for pumps and piping system.

The pumps selected consisted of one 400-gpm and two 850-gpm units. It was required that one unit discharge 400 gpm against a head of 37 ft and that the shut-off head be 43.5 ft while the other two pumps were each to deliver 850 gpm against a head of 37 ft with a shut-off head of 46 ft. From the head-capacity curves of pumps and system it is seen that with the 400-gpm unit in operation the rate of water delivery would be 560 gpm at a head slightly under 29 ft. A single 850-gpm unit would deliver 950 gpm to the system at a head of 32 ft. By operating a 400-gpm and an 850-gpm unit in parallel, 1,270 gpm would be delivered to the system against a total head of 36.5 ft, while with the two 850-gpm pumps operating in parallel 1,480 gpm would be delivered to the system against a head of 40 ft.

In reality the installation of the three pumps selected permits delivery of water to the system at four rates, namely, 560, 950, 1,270, and 1,480 gpm depending upon whether a 400-gpm, an 850-gpm, or a combination of these pump sizes is employed. The only means by which different quantities of water can be delivered to the system with this combination of pumps is either to increase or decrease the total head against which the pumps must operate. Thus, if it is desired to pump less than 560 gpm with the 400-gpm pump operating, it will be necessary to throttle the pump by partially closing the valve in the pump-discharge line to increase the friction head and thus decrease the flow.

The curves for head capacity of two pumps operating in parallel are determined by adding the quantity of water delivered by each pump at a given head condition.

In this connection it is interesting to note that many operators purchase pumps for the correct quantity of water to be delivered, but purposely increase the discharge head of the pump to be certain they obtain sufficient water. In many instances when this is done, owing to the characteristics of centrifugal pumps, the quantity of water pumped is considerably in excess of their requirements, an overload is put on the motor driving the pump, and the motor either overheats or burns out because of the overload.

Whenever a condition of this kind occurs, the load on the motor can be brought within its capacity by the simple expedient of closing the gate valve in the pump discharge slightly to increase the pump-discharge head. This decreases the rate of flow of water from the pump, decreases the load on the motor, and the excessive temperature in the motor windings immediately disappears.

138. Water Treatment.—Water treatment is generally required in conjunction with the cooling system in an internal-combustion-engine installation since most natural water supplies in the United States contain some dissolved mineral salts. The quantity of these salts per unit volume of water determines their suitability for use in cooling. In general the four mineral compounds found dissolved in natural-water supplies which cause the most difficulty are the following:

Calcium bicarbonate	$Ca(HCO_3)_2$
Magnesium bicarbonate	$Mg(HCO_3)_2$
Calcium sulfate	$CaSO_4$
Magnesium sulfate	$MgSO_4$

In addition to these four commonly found compounds, the chlorides and nitrates of both calcium and magnesium are sometimes present in the water supply. In rare instances, silicates of magnesium and calcium are found. When these

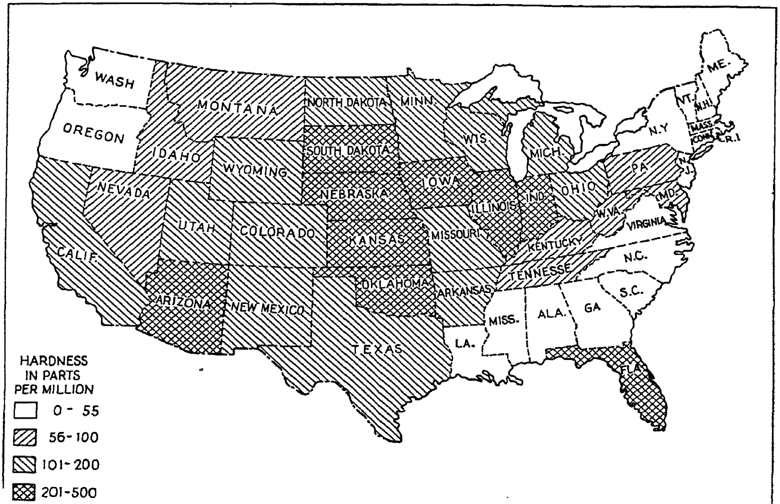


FIG. 105.—Average hardness, by states, of water furnished by public supply systems. (*United States Geological Survey data.*)

occur in any quantity, they may cause considerable difficulty in the scaling of engine jackets.

In some instances free oxygen is found in the water. The presence of bicarbonates always indicates the presence of free carbon dioxide. These gases, which are freed when the water is heated, cause corrosion of metal surfaces, and if steps are not taken to remove them they will cause severe pitting of engine jackets over a prolonged period of time.

The carbonates, sulfates, and silicates upon being heated deposit out in the form of scale. The carbonates of magnesium

and calcium form a relatively soft scale, while the sulfates form scales that are very hard and adhere closely to the metal. Silicate scales when formed are glassy hard and are extremely difficult to remove when they have once deposited in the cooling water passages of an engine.

If, for example, an open or single-circuit cooling system with a spray tower for cooling the water is used, the make-up water which must be added to the system to replenish that lost from the tower through evaporation and drift is constantly bringing more calcium and magnesium compounds into the cooling system. Since the hottest spot in the system is the surface of the cylinder liners and cylinder heads with which the water comes in contact, and since water upon being heated sufficiently will deposit out the mineral salts in solution, there is a constant depositing of the mineral salts in these locations. These deposits of scale act as an insulator and prevent the flow of heat through the metal walls of the cylinder and cylinder head into the cooling water. After a relatively short period of operation with such a system, it is necessary to shut down the engine and remove the scale deposits in order to secure satisfactory cooling of the engine.

An operating scheme of this character is entirely unsatisfactory, and if such an arrangement is to be used, all water entering the system must be treated to prevent this rapid accumulation of scale at those places in the engine most vitally affected by improper cooling.

Even when a double-circuit cooling system is employed, consideration must be given to the mineral content of the water used for the closed or jacket-water side of the system in order that too great a scale deposit is not produced over a long period of operation.

In general two methods of water treatment are being used, although in any particular instance it is always advisable to consult a reputable water chemist to ascertain the type of water treatment necessary. The treating method most widely used in internal-combustion-engine power plants employs zeolite, although in some instances a treatment using either lime or lime and soda ash is required.

The zeolite-treating method consists in using a complex, insoluble mineral compound consisting essentially of double silicates of aluminum, iron, or both. When containing a sodium

base, they possess the property of being capable of exchanging that sodium for the calcium and magnesium contained in hard water. After a sufficient quantity of water containing calcium and magnesium salts has been brought in contact with the zeolite to replace all the sodium with calcium and magnesium, the material is exhausted and the passing of additional hard water through the zeolite will result in no change in the hardness of the water. The zeolite can be brought back to its original condition, however, by treating it with a salt solution. The strong salt solution acts upon the zeolite to remove the calcium and magnesium and replace them with sodium in the zeolite. After this recharging operation has progressed to completion, the remainder of the salt solution together with the calcium and magnesium salts removed from the zeolite is drained to waste, the zeolite is washed, and the softener is again ready for its duty of removing calcium and magnesium from the water to be treated. This operation can be compared with the operation of an automobile storage battery. The softening of the water by the removal of calcium and magnesium and replacing them with sodium is comparable with starting an automobile from the storage battery. A prolonged starting period drains the battery down, and it is necessary to recharge it before it is capable of performing satisfactorily again.

The second method of softening consists in using either lime or lime and soda ash with the water to be treated in a suitably arranged tank in order that chemical reactions between the bicarbonates of magnesium and calcium and the lime take place and insoluble compounds are formed which precipitate. These reactions of lime and calcium and magnesium bicarbonates are shown by the following chemical equations:

$$\underset{\text{bicarbonate}}{\underset{\text{Calcium}}{Ca(HCO_3)_2}} + \underset{\text{lime}}{Ca(OH)_2} \rightarrow \underset{\text{(Precipitate)}}{\underset{\text{calcium carbonate}}{2CaCO_3}} + \underset{\text{water}}{H_2O}$$

$$\underset{\text{bicarbonate}}{\underset{\text{Magnesium}}{Mg(HCO_3)_2}} + \underset{\text{lime}}{2Ca(OH)_2} \rightarrow \underset{\text{hydroxide}}{\underset{\text{Magnesium}}{Mg(OH)_2}} + \underset{\text{carbonate}}{\underset{\text{calcium}}{2CaCO_3}} + \underset{\text{water}}{2H_2O}$$

(Precipitate out)

The first of these equations represents the lime softening in its simplest form. Both the chemical added and that which was already in the water are precipitated. In addition to the lime

shown by this equation, enough lime must be present to react with any carbon dioxide in the water and be itself precipitated as calcium carbonate. Calcium carbonate is slightly soluble, so that the reaction does not go to completion. There will usually be 25 to 40 ppm of calcium carbonate remaining in the treated water.

Examination of the second equation shown above reveals that twice as much lime is required for the removal of magnesium as for calcium. Furthermore, the reaction will not take place unless the water is caustic. Additional lime is required to produce this causticity, and after the removal of the precipitated magnesium hardness, the excess causticity must be neutralized with acid.

If the calcium and magnesium are combined in any form except the bicarbonate, they cannot be removed with lime alone. Soda ash is used, and the typical reaction with calcium sulfate is as follows:

$$\underset{\text{Calcium sulfate}}{CaSO_4} + \underset{\text{Soda ash}}{Na_2CO_3} \rightarrow \underset{\substack{\text{Calcium carbonate} \\ \text{(Precipitated)}}}{CaCO_3} + \underset{\substack{\text{Sodium sulfate} \\ \text{(Remains in solution)}}}{Na_2SO_4}$$

Following the lime or lime-soda ash treatment, the water should be filtered. Filtration is not required following the zeolite treatment.

The lime, the lime-soda ash, the zeolite, or the lime-zeolite type of treatment can each be adapted to either an open or a closed pressure-type system.

If the amount of water to be softened is small, it will probably be found that the pressure-type zeolite softener will be preferable in most internal-combustion-engine power plants because relatively little technical knowledge of water softening is required for its successful operation. There are some cases, however, where the zeolite treatment will not produce a satisfactory water, and in those instances other treating methods must of necessity be used.

Sodium hexametaphosphate [$(NaPO_3)_6$], marketed under the trade name of Calgon, which is being used in connection with municipal water-supply systems as a means for preventing the precipitation of calcium carbonate in water-distribution mains, has been found to be an effective chemical to add either in the

jacket water of an internal-combustion-engine installation or in the raw-water side of a double-circuit cooling system. When sodium hexametaphosphate is added to the water in a cooling system it prevents deposition of calcium carbonate in the engine jackets and effectively keeps this chemical in solution at elevated temperatures.

Where sodium hexametaphosphate is used as a treatment, either alone, or as an aftertreatment with zeolite-softened water, it is necessary to maintain a concentration of 2 to 5 ppm of the chemical in the water being circulated. It is also necessary to remove some water from the system, preferably at a constant rate, to keep the total mineral content of the water in the system below a predetermined concentration. This amount of removal is usually found to be about one-third of the water required for make-up purposes.

Thus in order to make effective use of this compound for conditioning engine-cooling water it is found necessary to keep a definite amount of the material in solution at all times, and in addition it is found necessary to limit the total amount of salts dissolved in the cooling system to a definite maximum by replacing a portion of the circulated water with fresh water. Experience in many instances with this material has proved it to be an excellent as well as a low-cost method of water treating.

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CHAPTER XIII

INTAKE AND EXHAUST SYSTEMS

The successful operation of an internal-combustion engine is so closely related to adequate and correct intake-air and exhaust-system construction that too much emphasis cannot be placed upon them. Intake air must be supplied to the engine free of foreign materials, which would cause damage or injurious wear to working parts, and in quantities sufficient for proper combustion of the fuel. The exhaust system must handle the discharged gases with a minimum of pressure drop and noise.

139. Air Requirements of Engines.—The quantity of air required for proper combustion and scavenging will be controlled by the type and design of engine. Although there is considerable variation in the quantity of air needed by the various types of engines, it is not particularly great when comparison is made between engines of a given type produced by different builders.

TABLE 35.—AIR REQUIRED BY VARIOUS TYPES OF DIESEL ENGINES

Type of engine	Air required per rated bhp	
	Lb per hr	Cfm at 60 F
4-stroke cycle mechanical-injection	10–13	2.2–2.8
4-stroke cycle air-injection	12–15	2.6–3.3
2-stroke cycle mechanical-injection, pump-scavenging.	22–23.5	4.8–5.1
2-stroke cycle air-injection	20–22	4.3–4.8
2-stroke cycle mechanical-injection crankcase-scavenging	18	3.9

NOTE.—For air volumes at other temperatures apply the following corrections:

Temperature.	70	80	90	100
Multiplier....	1.02	1.04	1.06	1.08

All volumes are calculated on the basis of dry air and a barometric pressure of 29.921 in. Hg.

This is shown in Table 35 giving the air requirements for various types of engines.

140. Air Filters.—Intake air for any internal-combustion engine should be cleaned prior to admission to the engine. Examination of carbon deposits on cylinder walls and piston-ring grooves of engines that have not been provided with air-cleaning equipment shows that over 40 per cent of it is foreign

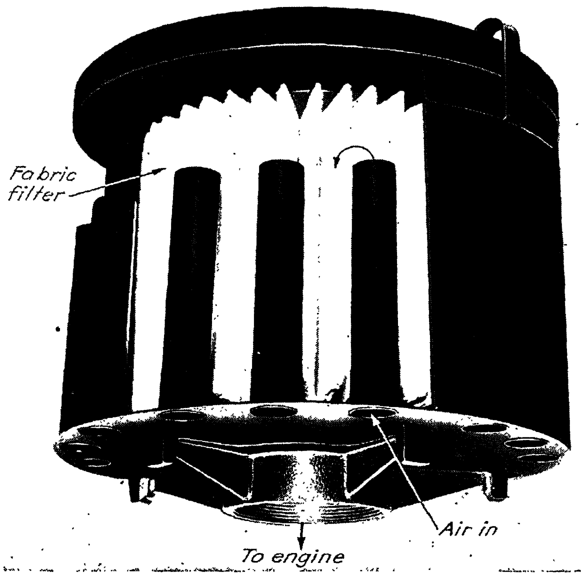


FIG. 106.—Dry-type filter designed for intake-air silencing. (*Courtesy of Staynew Filter Corp.*)

matter entering with the air. A considerable portion of this foreign material is silicates which, in the presence of the heat in the cylinder, fuse into glassy abrasive particles.

Air pollution varies from the high concentrations of several pounds per 1,000 cu ft in the desert sections of the western United States during severe dust storms to less than 0.2 grains per 1,000 cu ft in some areas. Even this latter relatively low dust concentration if not removed will, over a period of time, cause difficulties within the engine.

There are four types of filters available for removing dust from engine intake air classified as follows:

1. Dry type.
2. Viscous-impingement type.
3. Oil-bath type.
4. Electrostatic-precipitator type.

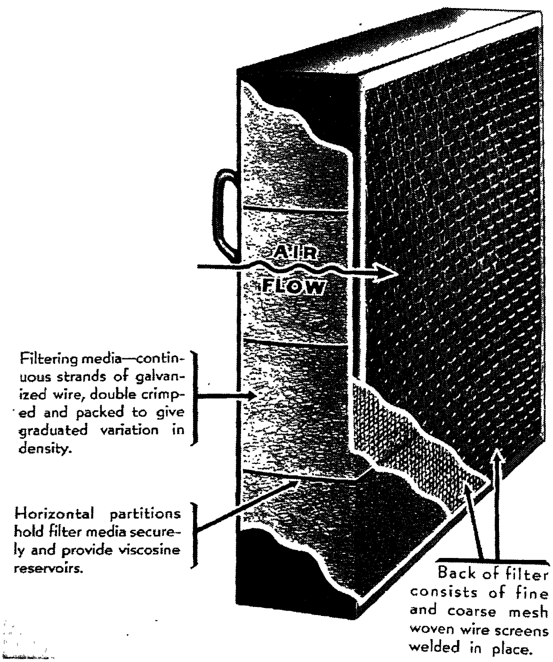


FIG. 107.—Viscous-type air-filter unit. (*Courtesy of American Air Filter Company.*)

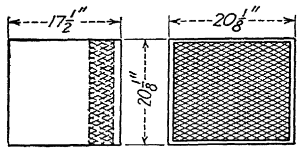
Dry-type filters are constructed of either fabric or spun-glass fiber through which the air is strained as it passes to the engine. The spun-glass type of filter has not been used in air-filtering service for internal-combustion engines to any extent. On the other hand, fabric filters employing a filtering type of fabric or felt as produced by Protectomotor and Coppus, Fig. 106,

have been used extensively with satisfactory results. This type of filter is cleaned by removing the filter element at frequent intervals and blowing off the accumulated dirt with compressed air, or cleaning with a vacuum cleaner.

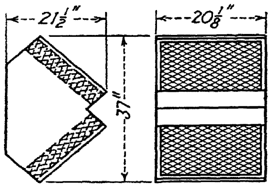
The *viscous-impingement* type of filter consists of a frame into which is packed stranded or crimped wire or a series of wire screens of graded mesh starting with a coarse-mesh screen on the air-inlet side and ending with a fine-mesh screen on the outlet side of the filter. Before being put into operation, filters of this type are dipped in a bath of oil to saturate the entire filter structure. Dust contained in the air passing through a unit of this type is caught and held by the oil adhering to the wire mesh or filler. When dust saturation is reached in 2 to 6 weeks under normal conditions, it is necessary to remove the filter element, clean it, and recharge it with oil. Air velocities through this type of filter should range from 350 to 450 fpm. The pressure drop through a clean filter for these velocities will vary from 0.25 to 0.375 in. of water. The efficiency of this type of filter is high and will remain so provided the filter units are cleaned regularly and the entire filter unit maintained in first-class condition.

In order to eliminate some of the cleaning difficulties inherent in this type of filter, a continuous automatic filter has been developed. This unit combines the cleaning and recharging operation within the filter housing. All filter elements are connected to a continuous chain so arranged that a filter cell can be cleaned and a freshly charged filter put into operation automatically without allowing any unfiltered air to get into the engine air-intake line. Automatic filters are usually not recommended for use with two-stroke-cycle pump-scavenging engines because the pulsating air flow causes excessive wear of the filter conveyor chains unless means are provided to eliminate effectively these pulsations at the filter.

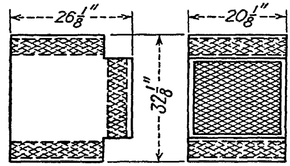
The *oil-bath* type of filter operates practically the reverse of the viscous-impingement type in that the air must pass through an oil spray which coats the dust particles and these in turn are trapped on a filtering medium. Thus instead of the dust striking a filter which is saturated with a viscous material, the dust particles are first coated with a viscous oil and then trapped by the dry filtering material. Heavy dust particles sink into the oil bath and do not reach the filter. Several makes of oil-



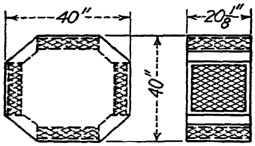
#1 OC-H
Max. flange top or base 6", back 12"



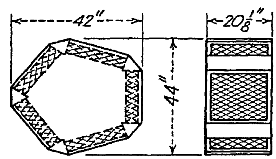
#2 OC-H
Max. flange top or base 8", back 12"



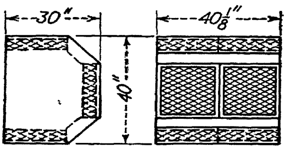
#3 OC-H
Max. flange top, base or back 12"



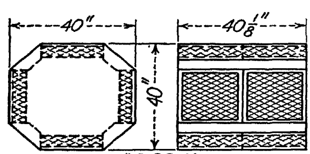
#4-OC-H
Max. flange top or base 20"



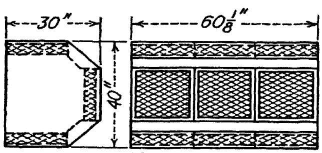
#5 OC-H
Max. flange top or base 20"



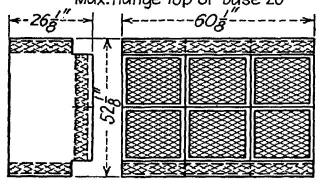
#6 OC-H
Max. flange top or base 16", back 20"



#8 OC-H
Max. flange top or base 20"



#9 OC-H
Max. flange top or base 16", back 20"



#12 OC-H
Max. flange top or base 12", back 30"

NOTE: The No. 10 OC-H Filter has the same cross section and takes

Courtesy of The American Air Filter Company.

TABLE 36.—CAPACITY OF VISCOUS-TYPE FILTER ASSEMBLIES

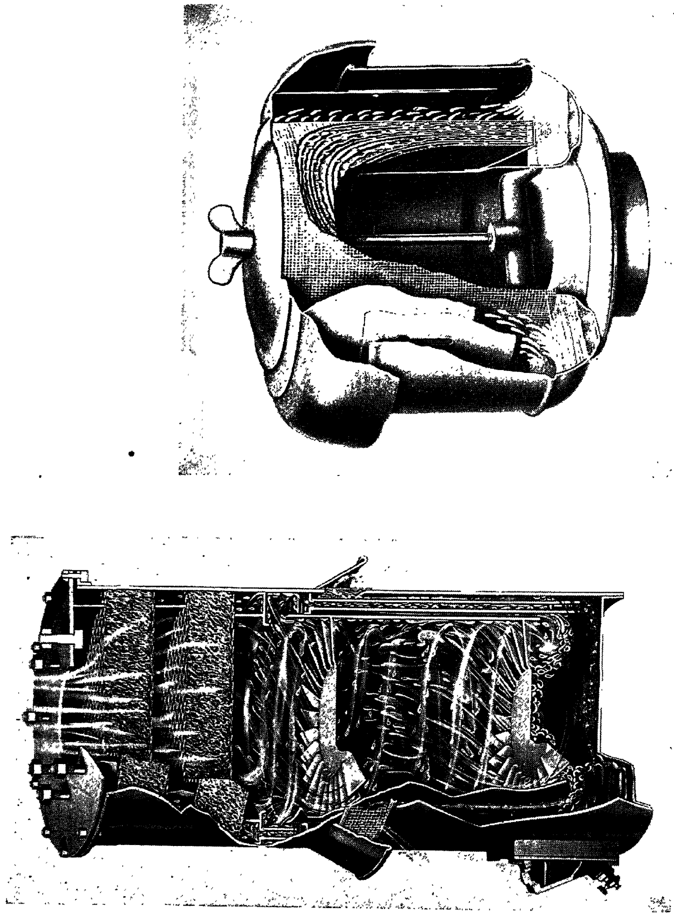
Type filter	Compressors: Piston displacement, cfm		4-cycle engines, hp				2-cycle engines crankcase scavenging, hp		2-cycle engines double-acting scavenging pump, hp
	Double acting	Single acting	1 cylinder	2 cylinder	3 cylinder	4 cylinder[b]	1 cylinder	2 cylinder[c]	
No. 5E	50	25	1–4	1–8	1–12	1–16	1–4	1–8	
No. 6E	85	45	5–7	9–14	13–21	17–28	5–7	9–14	
No. 8E	120	60	8–10	15–20	22–30	29–40	8–10	15–20	
No. 10E	200	100	11–16	21–32	31–48	41–64	11–16	21–32	
No. 12E	270	135	17–22	33–44	49–66	65–88	17–22	33–44	
No. 16E	500	250	23–38	45–76	67–114	89–152	23–38	45–76	
No. 1 OC-H	800	400	39–62	77–125	115–187	153–250	39–62	77–125	100–160
No. 2 OC-H	1,600	800	63–125	126–250	188–375	251–500	63–125	126–250	161–320
No. 3 OC-H	2,400	1,200	126–187	251–375	376–562	501–750	126–187	251–375	321–480
No. 4 OC-H	3,200	1,600	188–250	376–500	563–750	751–1,000	188–250	376–500	481–640
No. 5 OC-H	4,000	2,000	251–312	501–625	751–937	1,001–1,250	251–312	501–625	641–800
No. 6 OC-H	4,800	2,400	313–375	626–750	938–1,125	1,251–1,500	313–375	626–750	801–960
No. 7 OC-H[a]	5,600	2,800	376–437	751–875	1,126–1,312	1,501–1,750	376–437	751–875	961–1,120
No. 8 OC-H	6,400	3,200	438–500	876–1,000	1,313–1,500	1,751–2,000	438–500	876–1,000	1,121–1,280
No. 9 OC-H	7,200	3,600	501–562	1,001–1,125	1,501–1,688	2,001–2,250	501–562	1,001–1,125	1,281–1,440
No. 10 OC-H	8,000	4,000	563–625	1,126–1,250	1,689–1,875	2,251–2,500	563–625	1,126–1,250	1,441–1,600
No. 11 OC-H[a]	8,800	4,400	626–687	1,251–1,375	1,876–2,063	2,501–2,750	626–687	1,251–1,375	1,601–1,760
No. 12 OC-H	9,600	4,800	688–750	1,376–1,500	2,064–2,250	2,751–3,000	688–750	1,376–1,500	1,761–1,920

[a] The next larger size housing will be furnished with opening for one filter cell blanked off.

[b] Four cylinders or more.

[c] Two cylinders or more.

NOTE.—Performance shown above is the maximum permissible for each size filter. Most satisfactory results will be obtained by using the next larger size filter when the performance is close to the maximum indicated above.



B

A

FIG. 108.—Oil-bath-type filters. *A*. Cycoil type built by the American Air Filter Company. *B*. Type produced by Air Maze Corp.

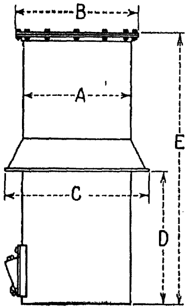
bath filters are on the market all operating on the same general principle. The major maintenance item is oil cleaning. Because of the accumulation of dirt in the oil it will sludge and must be replaced at frequent intervals, or the efficiency of the filter will be greatly reduced.

The *electrostatic precipitator*, which has been used extensively in industrial service and buildings for air cleaning and the removal of cinders and other injurious materials from stack gases, offers an interesting possibility for cleaning the air supply for engines. Development work with this type of air-cleaning equipment in recent years[1] has shown that it can be used satisfactorily for cleaning air entering residences, office buildings, and other air-conditioning installations. The friction loss of the air passing through the precipitator is extremely low and does not increase as the quantity of dirt which has been removed by the precipitator increases. The efficiency of this type of dirt-removal unit is extremely high and is not influenced materially by the increase in the quantity of dirt removed by the air passing through the unit.

While this unit is open to the objections that it is somewhat more expensive than the other types discussed and that it is necessary to employ high-voltage equipment for the efficient precipitation of the dirt from the air, the possibilities for the use of electrostatic precipitation for cleaning air for engines is excellent, and it will undoubtedly come into extensive use for this purpose in the near future.

141. Air-intake Muffling.—Disturbances have been caused in many instances by the resonant effects of pulsating intake-air flow. This is particularly true where the air intake is confined, such as in an alleyway between two buildings where the intake-air pulsations may create rather serious vibration problems. Objectionable vibrations which manifest themselves either in the structure housing the engines, in adjoining buildings, or in buildings some distance from the plant, may be caused by inadequate intake or exhaust muffling. It has been generally assumed that whenever anything in the neighborhood of an operating internal-combustion engine starts shaking it is the result of unbalanced mechanical forces within the engine transmitting vibrations

[1] PENNEY, G. W., A New Electrostatic Precipitator, *Elec. Eng.*, vol. 56, No. 1, p. 159, January, 1937.



DIMENSION TABLE

Type *W* Cycoil

Model No.	12W01	16W01	20W01	24W01	30W01	36W01	46W01
A, inches	12¼	16⅛	20¼	24¼	30⁵⁄₁₆	36⁵⁄₁₆	46¼
B, inches	14¼	18⅛	22¼	26¼	33⁵⁄₁₆	39⁵⁄₁₆	50⁵⁄₁₆
C, inches	17½	22	27½	32½	42	49	64½
D, inches	12⅝	14⅝	16⅝	19⅛	21¾	23⅝	30
E, inches	30¾	37⅛	43	48	56¼	65⅝	78
Maximum cfm	460	800	1,200	1,750	2,700	4,000	6,400
Maximum flange	6	10	12	16	22	26	36
Oil, gallons	2.2	3.5	6.0	8.5	13.8	19.8	32.4
Net weight	60	115	175	205	525	795	1,275
Shipping weight	90	140	235	330	720	1,020	1,570

Cycoil cleaners are furnished with studs and gaskets for connecting to intake-pipe flange.

CAPACITY TABLE

Application	12-in. Cycoil		16-in. Cycoil		20-in. Cycoil		24-in. Cycoil		30-in. Cycoil		36-in. Cycoil		46-in. Cycoil	
	Cfm piston displacement	Approx. hp	Cfm piston displacement	Approx. hp	Cfm piston displacement	Approx. hp	Cfm piston displacement	Approx. hp	Cfm piston displacement	Approx. hp	Cfm piston displacement	Approx. hp	Cfm piston displacement	Approx. hp
Four-cycle engines:														
1-cylinder	115	36	200	60	300	95	435	140	675	215	1,000	315	1,600	500
2-cylinder	230	72	400	125	600	185	875	275	1,350	415	2,000	615	3,200	1,000
3-cylinder	345	108	600	185	900	280	1,310	415	2,025	630	3,000	930	4,800	1,500
4 (or more)-cylinder	460	145	800	250	1,200	375	1,750	550	2,700	840	4,000	1,250	6,400	2,000
Two-cycle engines crankcase scavenged:														
1-cylinder	230	39	400	68	600	100	875	148	1,350	225	2,000	330	3,200	530
2 (or more)-cylinder	460	78	800	136	1,200	200	1,750	296	2,700	450	4,000	660	6,400	1,060
Two-cycle engines with scavenger blower	460	75	800	130	1,200	195	1,750	280	2,700	440	4,000	650	6,400	1,040

FIG. 109.—Dimensions and capacities of Cycoil air filters. (*Courtesy of American Air Filter Company.*)

through the soil. In too many instances this is far from the correct interpretation of the cause of the vibration annoyance, which may be due to pulsating air waves resulting from the inadequate muffling of air intake or exhaust or both.

Since the air intake for an internal-combustion engine can be the source of considerable annoyance to property owners in the neighborhood of the engine, it is advisable to give serious consideration to providing mufflers for air intakes where such annoyance might be created. Muffling of the air intake for four-stroke-cycle engines is relatively simple and may be effectively accomplished by the use of an air filter having some muffling properties. There are at least two such filters on the market. Muffling the air intake of two-stroke-cycle engines, and particularly those engines employing pump scavenging, is considerably more difficult. Consequently suitable designed mufflers must be used with these engines wherever intake-air noises must be eliminated. Two-stroke-cycle engines using Roots type blowers for scavenging may produce a very objectionable high-pitched tone. The air intake to a Buchi supercharger or other high-speed fan often produces an objectionable whine.

Each intake-muffling problem must be given consideration in terms of the conditions surrounding the particular case. No specific rules can be set forth, but if it appears at all probable that objectionable disturbances may result from intake-air noises, the designer of the installation should provide muffling equipment for the air intakes. In any event it is advisable to discuss the problem with specialists in the field of acoustical muffling in order that a rational solution to the problem may be reached.

There are several things that can be done to eliminate objectionable air-intake noises. The installation of a suitable intake-air muffler is the most satisfactory. Providing a large plenum chamber between the air filters and the air-inlet pipe to the engine will tend to reduce pulsations at the air filter. Bringing the air into the filter housing from a high level as shown in Fig. 114 is oftentimes very effective. The installation of a Venturi nozzle in the entrance of the inlet-air pipe to the engine has also helped to reduce noise.

142. Intake-air Pipe Size.—The size of the intake-air line for any engine is determined by the pressure drop resulting from the flow of air through the line. This drop in air pressure should not be more than 5 to 5.5 in. of water, or approximately 0.2 psi. A pressure of 1 in. of water equals 0.0361 psi. For purposes of quick conversion from inches of water to pounds per square inch the following table will be of considerable value.

TABLE 37.—CONVERSION FROM INCHES OF WATER TO POUNDS PER SQUARE INCH

Inches water	Psi
1	0.036
2	0.072
3	0.108
4	0.145
5	0.181
6	0.217
7	0.253
8	0.289
9	0.325

For air under pressures less than 2 psi gauge, and where the pressure drops are less than 10 in. of water, only a slight error is introduced into the calculations if the formulas normally applied to the flow of water are used. The basic equation of flow is

$$V_s = \sqrt{2gH} \tag{32}$$

where V_s = velocity, feet per second.
g = acceleration due to gravity, or 32.2 ft per sec per sec.
H = head in feet of air causing flow.

When the velocity is given in feet per minute, Eq. (32) becomes

$$V = 1096.5 \sqrt{\frac{h}{W}} \tag{33}$$

where V = velocity, feet per minute.
h = head or pressure, inches of water.
W = weight of air, pounds per cubic foot.

For air at 70 F with a barometric pressure of 29.92 in., Eq. (33) becomes

$$V = 4{,}005 \sqrt{h} \tag{34}$$

Thus the air velocity for a pressure or head of 1 in. of water at standard conditions and neglecting friction would be 4,005 fpm.

Further reference will be made to Eq. (34) in connection with determining pressure loss in pipe entrances and pipe bends.

Two types of loss in pressure occur when air is required to move in a confined space. They are frictional loss from contact of the air with the walls of the pipe or duct, and the loss resulting from change in direction of air flow or change in size of the duct

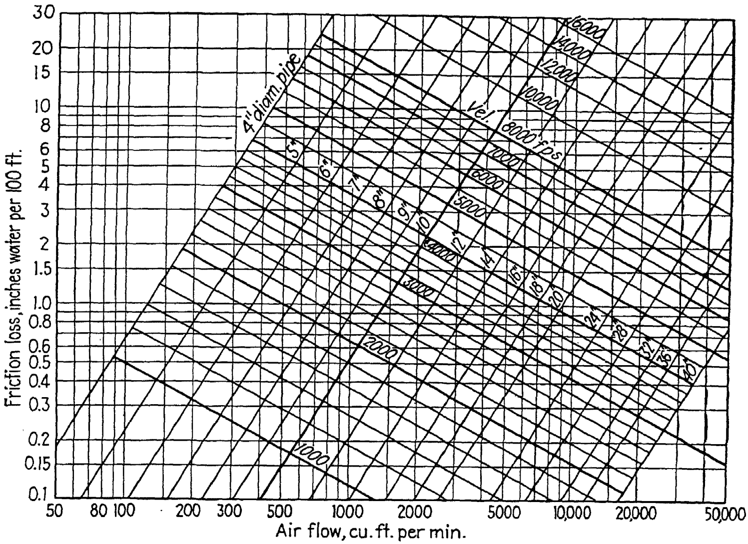


FIG. 110.—Friction loss of air flowing through circular pipes. (*Courtesy of Buffalo Forge Company.*)

creating a change in air velocity. The loss in pressure due to air flow in straight pipe has been calculated by means of various formulas all of which are more or less empirical and are based upon experimental data. The simplest of these which gives fairly accurate results is

$$h = \frac{KV_s^2}{D} \tag{35}$$

where h = pressure drop or head, inches of water per 100 ft.

V_s = velocity of air flow, feet per second.

D = diameter of duct, feet.

K = constant, varying from 0.00055 to 0.000635.

Pressure drop resulting from air flow in round ducts and pipe is readily determined by means of a chart such as Fig. 110 which gives pipe size, air velocity, and friction loss for various quantities of air flow. This chart, based upon an empirical formula somewhat different from Eq. (35), gives friction losses slightly less than those calculated by means of this equation.

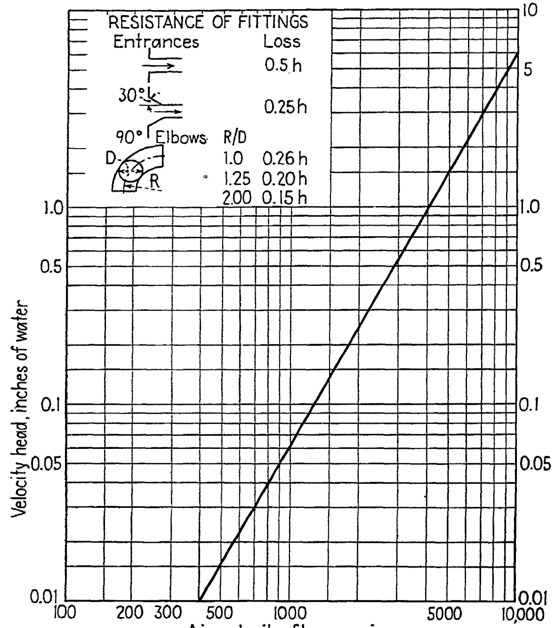


FIG. 111.—Air resistance of entrances and bends.

In addition to the friction loss in straight runs of pipe, losses occur in pipe bends and at points where the pipe changes size. Thus a change in size of pipe occurs where the air-intake line connects to the air-filter housing or plenum chamber. Pressure losses occurring in bends or at points of change in pipe size or pipe entrances are generally measured in "velocity heads."

As shown previously by means of Eq. (34), an air pressure of 1 in. of water will create an air velocity of 4,005 fpm neglecting

friction. Equation (34) can be rewritten in the form

$$h = \left(\frac{V}{4{,}005}\right)^2 \tag{36}$$

In this form h is ordinarily referred to in hydraulics as *velocity head* since it is the pressure or head which creates the velocity V. Figure 111 gives values of h for velocities ranging from 400 to 10,000 fps.

143. Calculating Air Intake Sizes.—For the purpose of illustrating the application of the formulas and charts developed, consider that the air-intake line for a 500-hp four-stroke-cycle mechanical-injection engine is 40 ft long and contains one 90-deg. bend with an R/D ratio of 1 and an abrupt change in the size of the line where it connects to the air-filter housing. It is desired to keep the pressure drop through the line at approximately 3 in. of water. From Table 35 the engine will require 2.5 cfm of air per brake-horsepower at 60 F. The total air requirement per minute for the engine when the air temperature is 70 F becomes $2.5 \times 500 \times 1.02 = 1{,}280$ cfm. From Fig. 110 for an air flow of 1,280 cfm when using a 7 in. diameter pipe, the air velocity is 4,700 fpm and the pressure loss per 100 ft is 4.9 in. of water. From Fig. 111 for a velocity of 4,700 fpm, the velocity head h becomes 1.35 in. of water and the loss through the 90-deg. elbow and the entrance is

$$1.35 \times (0.5 + 0.26) = 1.02 \text{ in. water}$$

The total loss through the pipe and fittings becomes

$$(0.4 \times 4.9) + 1.02 = 2.98 \text{ in. of water}$$

A pressure drop of 3 in. was desired, and it appears that a 7 in. diameter pipe will suffice. Since this is not a pipe size normally carried in stock, an 8 in. diameter pipe would be used because it is the nearest standard pipe size available in which the allowable pressure drop would not be exceeded.

In selecting the size of an air-intake line it is often desirable to make a series of calculations for various pipe sizes in order to determine the pressure drop for each size and from these calculations determine that pipe diameter which best suits the conditions involved. Such a calculation is set up in Table 38 for the problem just illustrated, using four different pipe diameters.

TABLE 38.—CALCULATION OF PRESSURE DROP THROUGH PIPE

Diameter of pipe, inches	7	8		10
Air velocity, feet per minute	4,700	3,700	2,900	2,300
Friction loss per 100 ft, inches	4.9	2.5	1.4	0.82
Loss through 40-ft pipe, inches	1.96	1.00	0.56	0.33
Velocity loss through fittings, 0.76h	1.02	0.64	0.40	0.25
Total loss through air intake, inches.		1.64	0.96	0.58

These calculations have assumed that the air was flowing through the intake line at constant velocity. This condition is never strictly true, although in the case of two-stroke-cycle engines using blower scavenging or four-stroke-cycle engines

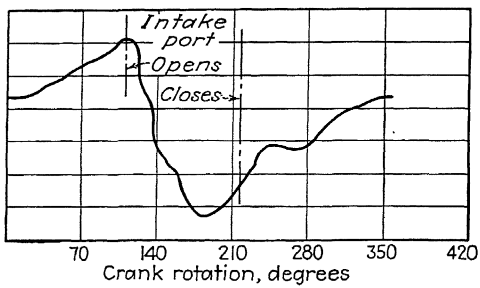


FIG. 112.—Variation of intake-air pressure in two-stroke-cycle, single-cylinder, blower-scavenged diesel operating at 1,200 rpm. (*Courtesy of Burgess Battery Company.*)

having six or more cylinders it is sufficiently close to actual conditions. When two-stroke-cycle pump-scavenging engines are considered, as well as either two- or four-stroke-cycle engines having less than six cylinders, there is a considerable variation in the velocity of the air traveling through the intake-air line unless provisions have been made to iron out these irregularities through the installation of a large air plenum chamber immediately adjacent to the engine into which the intake-air line feeds. This variation for a single-cylinder engine is shown in Fig. 112.

144. Location of Air Intakes.—The location of an air intake for an internal-combustion engine is determined partly by the design of the structure required for housing the power-plant equipment and partly by the limitations imposed by the length and diameter of air-intake line required. The air intake may

be located near the ground and adjacent to the power-plant building, in the building wall, on the roof, or in some instances within the building proper, Fig. 113. The latter location, inside the power plant, should not be used unless absolutely necessary to satisfy special design requirements, since it takes the air for fuel combustion from inside of the power-plant build-

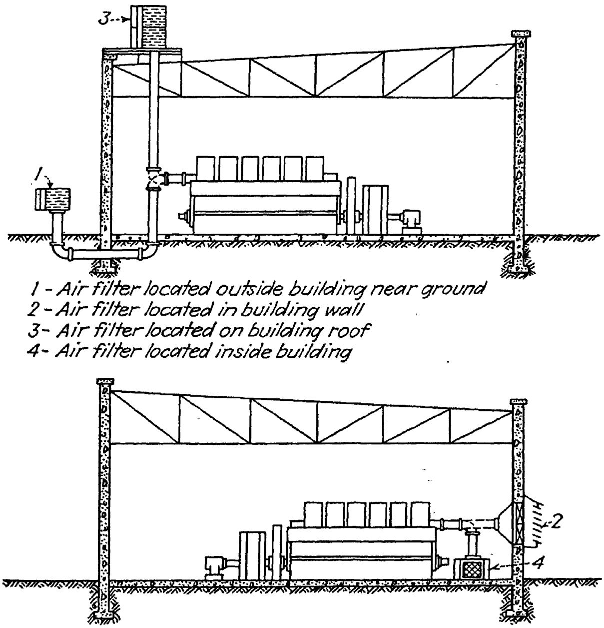


Fig. 113.—Typical air-filter and air-intake locations.

ing and makes it difficult to heat the building adequately in cold weather.

When the air filters and air intake are located on the roof of the building, cleaner air will be obtained in most instances and the amount of filter cleaning will consequently be less than when the filters are located near the ground. If the air filters are located on the building roof, they must be made readily accessible for cleaning; otherwise they will be neglected and poor engine operation and loss of power will result.

In general, air filters will be found to fit best into the general plant design and operation when they are located either adjacent to the building at some slight elevation aboveground, in the building wall, or in a housing attached to the building wall. One arrangement of housing providing support for the exhaust muffler as well as location for the air filters is shown in Fig. 114.

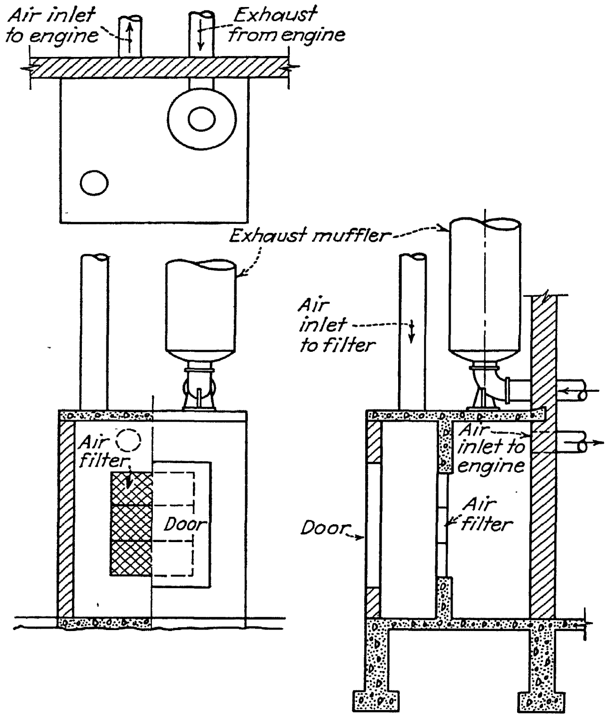


FIG. 114.—Combined air-filter housing and support for exhaust muffler.

While the foregoing discussion of air-intake and air-filter location has been based primarily upon the use of dry or viscous-impingement types of filters, the same general remarks will apply, with slight modifications, to other types of air-filter construction.

The location of air filters and exhaust mufflers on the roof deck of a low-roofed addition to the plant may be advantageous in some cases, Fig. 115, although suitable provisions for cleaning and maintaining the filters must be provided.

145. Don'ts for Air-intake Construction.—Experience with the design of many air intakes has shown that the following should be avoided:

1. Don't install air intakes inside the engine room except for very small engines or in a territory where building heating need not be considered.

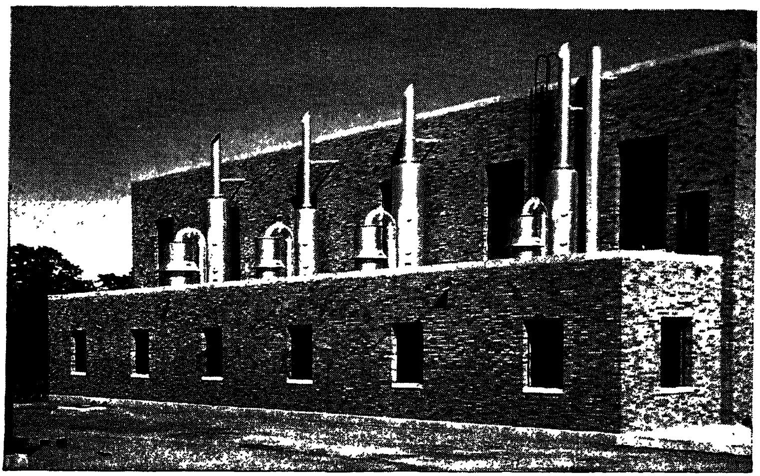


FIG. 115.—Air filters and exhaust snubbers mounted on roof of plant shop. Access is gained by a stairway inside the plant.

2. Don't install air-intake filters too close to the roof of an engine room or excessive vibration of the roof may result from pulsating air flow through the filters.

3. Don't install a pump-scavenging or air-injection engine without providing a large plenum chamber in the air intake between the air filters and the engine. The failure to provide a sufficiently large plenum chamber may result in rather annoying and perhaps serious vibrations.

4. Don't take air from a confined space. If this is done, serious vibration problems resulting from air pulsations may ensue.

5. Don't locate air-intake filters in an inaccessible spot.

6. Don't use an air-intake line with too small a diameter or make it too long. Engines can be starved for air, and air is just as necessary for an engine to produce power as is the fuel.

146. Checking Air Filters.—Checking of the condition of air filters should be done periodically to ensure that the engines will obtain sufficient combustion air at all times. In order that this may be done systematically and accurately, an arrangement utilizing a simple U-tube pressure gauge, Fig. 116, will be found very convenient. As the filter becomes clogged with dirt, the

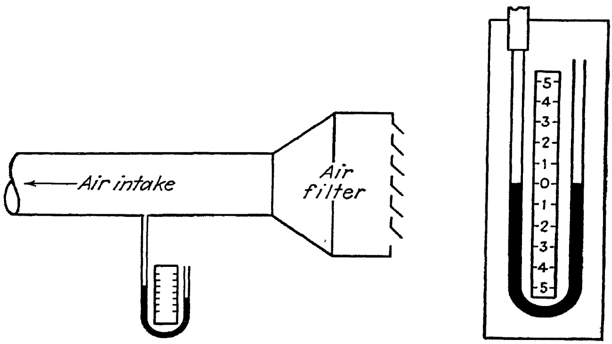


FIG. 116.—U-tube gauge for determining cleanliness of air filters. On the detail of the gauge each division above or below zero is just one-half the value given. Thus for measuring differential pressures in inches of water, the distance from 1 above 0 to 1 below 0 would be 1 in., and the distance from 5 above to 5 below 0 would be 5 in.

suction pressure measured by the U tube will increase. When this suction pressure reaches approximately 10 to 12 in. in most cases, the filters should be cleaned. This simple gauge can also be used to check the efficiency of the filter cleaning.

147. Engine-exhaust Behavior.—Engine-exhaust behavior has been the subject of considerable experimental investigation in recent years. These investigations show that pressures as great as 10 psi are built up in the exhaust pipe within 0.003 sec after the engine exhaust opens, Fig. 117. Such pressure rises result in very rapid acceleration rates as the gas moves out the exhaust line and the gas velocities attained are high. The pressure drop following is practically as rapid as the initial pressure rise.

As pointed out by one investigator,[1]

The gas in an engine cylinder is still under considerable pressure and temperature at the bottom of the piston's power stroke. *Immediately following the opening of the exhaust valve or port, this gas is forcibly ejected down a pipe like a compressed spring suddenly released.* Like the spring, the gas possesses mass and elasticity, and when released and the valve is again closed, it overruns, that is, stretches itself out well beyond its normal length. This, coupled with the cooling of the gas and condensing of the vapors, results in a partial vacuum following the "slug."

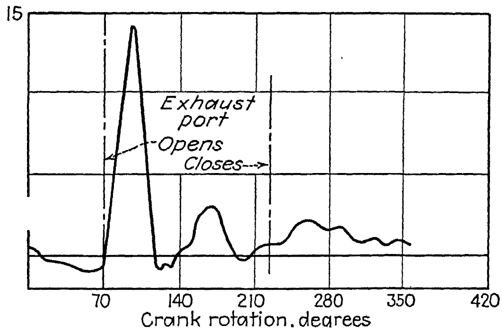


FIG. 117.—Variation in exhaust pressure in two-stroke-cycle, single-cylinder, blower-scavenged diesel operating at 1,200 rpm. (*Courtesy of Burgess Battery Company.*)

With the slug out of the way, air is sucked into the pipe and there is set up in the exhaust conduit a severe oscillating condition. Pressure waves of considerable amplitude are reflected back and forth in the pipe line at acoustic velocities. The frequency of the reflected wave depends primarily on pipe length and somewhat on pipe diameter and gas temperature. In any event, this condition is detrimental since it results in noise, and quite often, impaired engine performance. For this reason, it is necessary to process the exhaust waste before it is released to atmosphere to entirely prevent the destructive reverberation.

The characteristics of the pressure oscillations resulting from this expulsion of the gas slug without muffling are influenced materially by the length of the exhaust line connected to the engine, Fig. 118. It is well known that when a muffler is not used the length of the exhaust pipe can be proportioned to have

[1] CHIPLEY, ALFRED S., Diesel Engine Exhaust Silencing, *Eng. Bull.* 117, Burgess Battery Company, Chicago, 1941.

either a beneficial or harmful effect upon the scavenging of exhaust gases. From Fig. 118, it is seen that only a relatively slight change in the exhaust-pipe length creates a cylinder pressure at the time of exhaust valve closing either above or

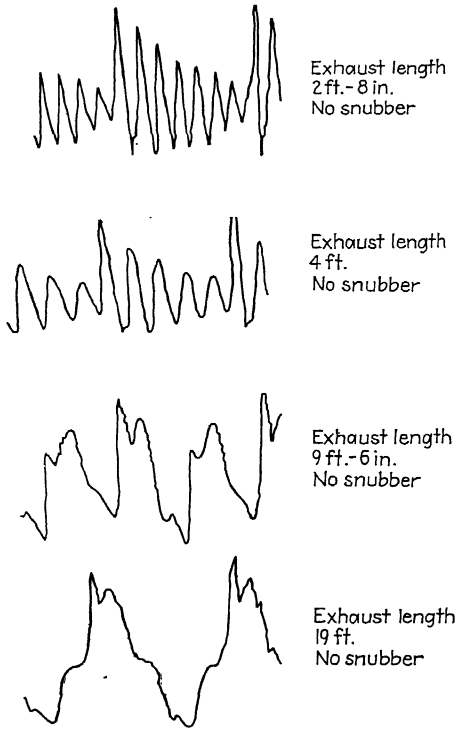


FIG. 118.—Effect of exhaust-pipe length on character of exhaust pulsations. (*Courtesy of Burgess Battery Company.*)

below atmospheric pressure. Either condition might be harmful. An excessively high cylinder pressure may reduce the quantity of oxygen in the cylinder sufficiently to interfere with proper fuel combustion.

The oscillating characteristics of the exhaust have a very marked effect upon the fuel consumption, exhaust back pressure, and exhaust temperature of an engine, Fig. 119. In this particular series of tests the effect of different lengths of exhaust pipe, with or without a suitable muffler attached, on fuel consumption, exhaust back pressure, and exhaust temperature was studied.

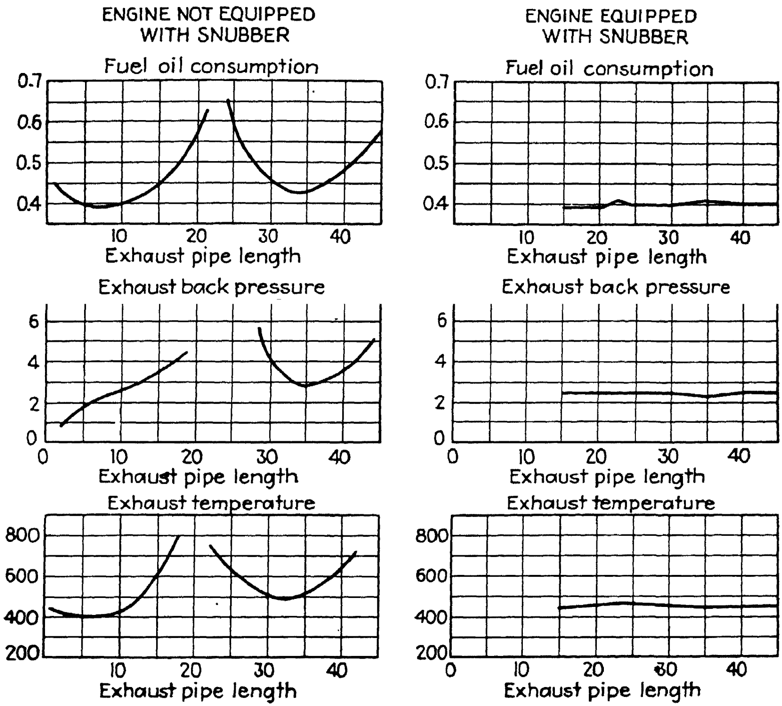
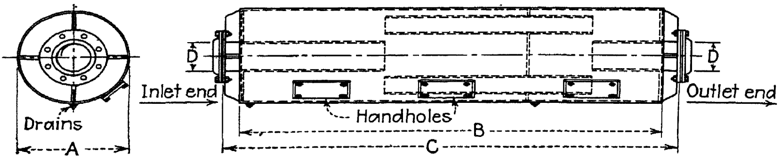


FIG. 119.—Effect of exhaust snubber on fuel consumption, exhaust back pressure, and exhaust temperature. (*Courtesy of Burgess Battery Company.*)

The load on the engine was maintained constant. With no muffler on the engine, exhaust pipe lengths of approximately 8 and 32 ft showed the points of lowest fuel-oil consumption per brake horsepower-hour as well as the lowest exhaust temperatures. On the other hand, the length of exhaust line showed little if any influence on fuel-oil consumption, exhaust back



DIMENSIONS IN INCHES

Size	A	B	C	D	Estimated weight, lb
No. 2	5⅛	19¼	Pipe threads	1	12
No. 2½	6⅛	23¼	Pipe threads	1¼	18
No. 3	7⅛	25¼	Pipe threads	1½	25
No. 4	8⅛	31¼	Pipe threads	2	50
No. 5	10⅛	36	42	2½	70
No. 6	12⅛	42	48	3	100
No. 7	13¼	46	52	3½	120
No. 8	14¼	48	54	4	180
No. 10	18¼	58	64	5	280
No. 12	20¼	72	78	6	380
No. 16	24¼	87	93	8	640
No. 20	32⅜	97	104	10	1,200
No. 24	38⅜	117	124	12	1,750
No. 28	42⅜	132	139	14	2,100
No. 32	48⅜	140	147	16	3,230
No. 36	50⅜	150	158	18	3,840
No. 40	54⅜	158	166	20	4,400
No. 44	60½	167½	175½	22	6,200
No. 48	66½	182½	190½	24	7,400
No. 52	72	197½	205½	26	8,700
No. 56	78	213	221	28	10,500
No. 60	83	228	236	30	12,300

FIG. 120.—Standard sizes of Maxim, model MUC silencers. (*Courtesy of The Maxim Silencer Company.*)

pressure, and exhaust temperature when equipped with a suitable muffler.

148. Exhaust-system Functions.—The operation of the exhaust system of an internal-combustion engine involves three separate activities, all independent, yet all necessary. The system must carry the exhaust gas from the engine to a point where its rejection will not be objectionable, release it with a minimum of noise, and impose a minimum back pressure on the engine exhaust. Providing an adequate exhaust system is not easy since it involves consideration of the hydraulics and acoustics of a compressible fluid under pulsating flow.

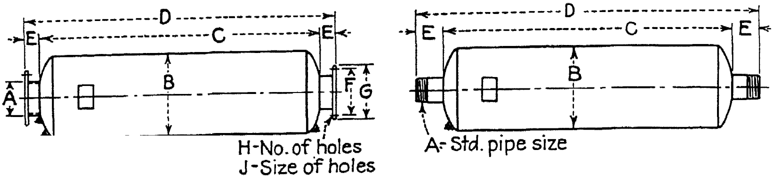
The problems involved are not subject to rigorous mathematical treatment. While the progress that has been made in the development of muffling equipment has been exceptional, most of it has come about through endless experimentation. This experimental work has evolved several very excellent exhaust mufflers, and today the plant designer can select that muffling equipment which best suits the conditions with which he is confronted.

149. Mufflers.—Mufflers suitable for use with internal-combustion engines in stationary service are produced by two well-known firms in the United States. In some cases, mufflers are also constructed by the engine manufacturer. Most mufflers are now fabricated of welded steel plate, although some are made of cast iron. There are several types available including the standard or conventionally designed unit, spark-arresting type for hazardous locations, and a combination muffler and waste-heat boiler.

The muffler size is determined by the cylinder dimensions and type of engine. The correct size unit for a particular engine can be obtained from either the engine or muffler manufacturer.

150. Exhaust Pipe Size.[1]—The size and arrangement of exhaust piping from a diesel engine should be such that it will not cause undesirably high back pressures in the engine cylinders. High back pressures reduce the capacity and efficiency of the engine, and, indirectly, increase lubrication difficulties and maintenance expense, by inducing high cylinder temperatures.

[1] The material presented here was prepared especially for the author by Max Rotter, who, prior to his death in 1940, was consulting engineer for Busch-Sulzer Bros. Diesel Engine Company.



APPROXIMATE DIMENSIONS IN INCHES. WEIGHT IN POUNDS

Catalogue No.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*J*	Net weight
STC	1	4½	18	22	2	....	....	..	...	9
STC 1½	1½	6½	20	24	2	....	....	..	...	13
STC 2	2	8	26	32	3	....	....	..	...	24
STC 2½	2½	10	36	42	3	....	....	...	...	46
STC 3	3	12	40	48	4	....	....	..	...	85
STC 3½	3½	14	46	54	4	....	....	..	...	108
STC 4	4	14	58	66	4	....	....	..	...	142[a]
STC 5	5	16	71	77	3	8½	10	8	⅞	193[a]
STC 6	6	18	90	96	3	9½	11	8	⅞	240[a]
STC 8	8	24	102	108	3	11¾	13½	8	⅞	550[b]
STC 10	10	30	128	134	3	14¼	16	12	1	960[c]
STC 12	12	36	136	142	3	17	19	12	1	1,525[d]
STC 14	14	36	190	198	4	18¾	21	12	1⅛	2,175[d]
STC 16	16	42	204	212	4	21¼	23½	16	1⅛	2,900[d]
STC 18	18	48	204	212	4	22¾	25	16	1¼	3,700[d]
STC 20	20	54	218	226	4	25	27½	20	1¼	4,850[d]
STC 22	22	60	220	230	5	27¼	29½	20	1⅜	5,980[d]
STC 24	24	60	256	266	5	29½	32	20	1⅜	7,400[e]
STC 26	26	66	270	280	5	31¾	34¼	24	1⅜	9,500[e]
STC 28	28	72	272	282	5	34	36½	28	1⅜	11,600[e]
STC 30	30	72	308	318	5	36	38¾	28	1⅜	13,800[e]
STC 32	32	78	308	318	5	38½	41¾	28	1⅝	16,000[e]

[a] Add 30 lb for handholes
[b] Add 70 lb for handholes
[c] Add 90 lb for handholes
[d] Add 135 lb for handholes
[e] Add 240 lb for handholes

FIG. 121.—Standard sizes of Burgess STC series exhaust snubbers. (*Courtesy of Burgess Battery Company.*)

The total back pressure should not exceed 15 in. of water at the outlet of the engine exhaust header, and will be built up by the friction in the exhaust pipe and the resistance in the exhaust silencer and spark arrester. The former should not exceed 3 in. of water; the latter combined, 12 in. of water.

Exhaust piping should be short and straight as possible. Bends should be of long radius. Abrupt changes in diameter should be avoided. In determining the pipe size, allowance must be made for the fouling of the piping. To avoid long and involved calculations,

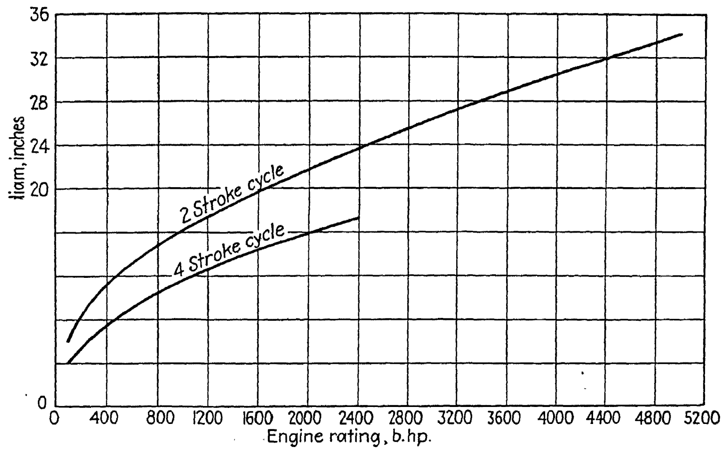


FIG. 122.—Exhaust pipe sizes for diesel engines. (*Data prepared by Max Rotter.*)

the appended curves, Fig. 122, have been prepared as applying to an average arrangement of piping to impose a back pressure of about 3 in. of water in the piping alone. These curves represent modern good practice when the piping is not unusually long or crooked, or conversely, unusually short and straight.

Piping arrangements for several typical exhaust systems are discussed in Chap. XIV, Art. 158.

151. Engine-exhaust Design.—The design of an exhaust system for any engine involves several fundamentals which must be followed if a satisfactory system is to result. These rules are as follows:

1. Use a pipe with a diameter equal to or exceeding the values given by Fig. 122.

2. Make the exhaust line as short as possible.
3. Keep the number of bends in the line to a minimum.
4. Make provisions for adequate expansion in the exhaust line.
5. The tail pipe on the muffler should extend above the building roof, and the outlet end should be beveled at an angle of 45 deg. or more.

Some engines are equipped with individual exhaust pipes for each cylinder which terminate below the bottom of the engine frame. It is usual to build a rectangular concrete conduit beside the engine foundation under the engine-room floor into which these individual exhaust pipes discharge. Where such a concrete duct is installed, provisions must be made to allow for expansion of the duct, or serious damage may be caused to the engine and generator foundation block.

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———: Correct Exhaust Piping Improves Engine Performance, *Power*, vol. 75, No. 11, p. 401, Mar. 15, 1932.

CHAPTER XIV

PIPING SYSTEMS

Piping design for an internal-combustion-engine power plant deals with the arrangement of pipe, valves, and accessories for the successful operation of the fuel and lubricating-oil systems, cooling-water system, starting-air system, and intake and exhaust systems. Hydraulic calculations necessary to determine pressure losses and pipe sizes for the various fluids handled are discussed elsewhere. In this chapter the mechanical and structural problems involved in the layout of the plant piping and its operation are discussed.

152. Piping-system Function.—Any well-designed piping system must satisfy the following requirements:

1. It must function properly and efficiently in conjunction with the mechanical equipment it serves.
2. It must be arranged in a manner to facilitate operation and maintenance.
3. It must be as simple as possible and still perform its primary functions.
4. It must be designed as a coordinated and symmetrical part of the entire plant layout.

Good piping design requires that careful and detailed consideration be given to all its component parts in order that the above requirements can be realized. Care taken in developing and planning the plant piping will assure an economical installation as well as one that can be easily operated and maintained.

153. Plan the Piping Assembly.—Before making any attempt to install piping materials, have a comprehensive plan of it prepared. This requires that a layout of all the pipe, fittings, and valves be made to scale in order that conflicts in location of pipes be avoided and a suitable arrangement for all piping be provided. Too often the piping up of a plant is not considered until all equipment is installed and the pipe and fittings are on the job. Under such conditions, it is little wonder that the piping is generally a cut-and-compromise affair.

Systematic planning prior to installation eliminates these difficulties. Piping drawings may be simple one-line diagrams,

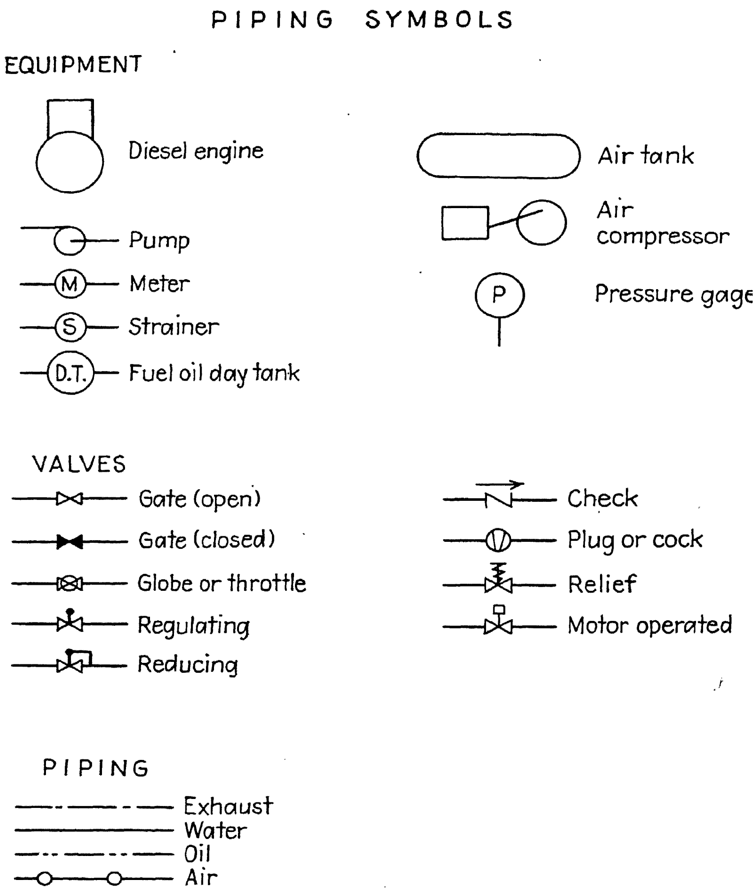


FIG. 123.—Symbols used for indicating piping and piping accessories.

either in plan and elevation or in simple mechanical perspective. Samples of such diagrams with dimensions are shown in Fig. 124.

These sketches are necessary to obtain a correct bill of material for the job, to eliminate conflicts in the location of the numerous piping runs, and to facilitate hydraulic calculations. They may be extremely simple when only a single engine of small horsepower capacity is involved, or they may become very complex in the case of installations involving many engines of large horsepower capacity.

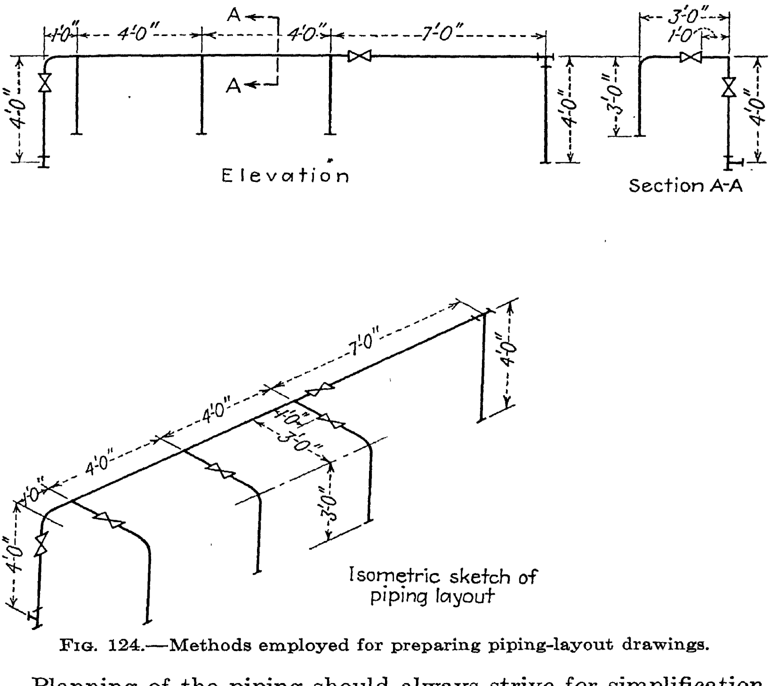


FIG. 124.—Methods employed for preparing piping-layout drawings.

Planning of the piping should always strive for simplification insofar as possible. Needless fittings, bends, valves, and other accessories should be eliminated in order to reduce frictional losses, decrease the number of joints where leaks might occur, and save on future maintenance of the piping system.

154. Cooling-water Piping.—The engine cooling-water system is an extremely important part of the plant. All engines now used for stationary power purposes are liquid cooled, and a

continuous supply of water must be passing through the engine-cooling passages in order for it to operate. Water must be delivered to the engine continuously in sufficient quantity while the engine is in operation.

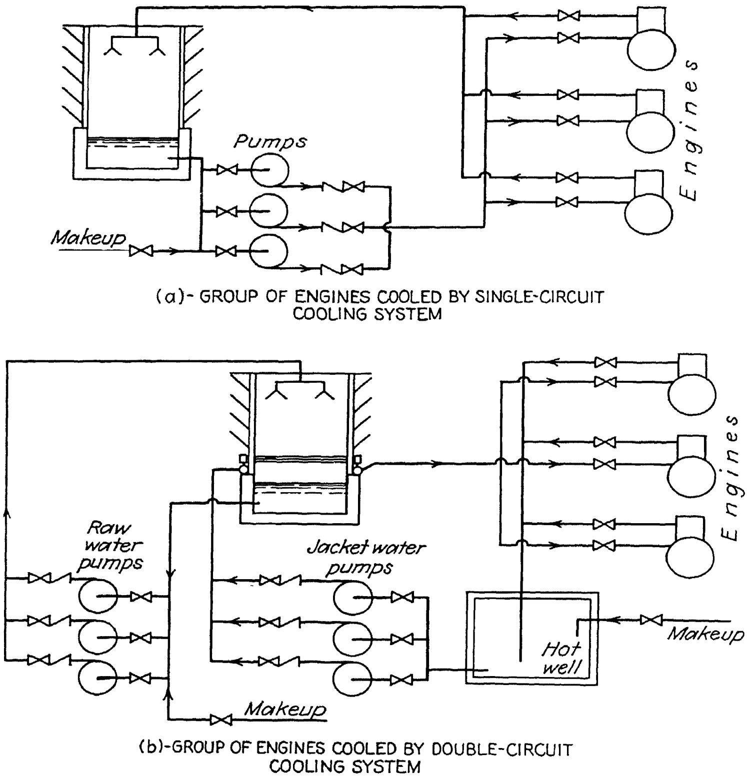


FIG. 125.—Typical cooling-water systems.

In order that the correct quantity of water can be delivered without interruption, care must be taken to ensure that the piping layout connecting the engine, water pumps, and cooling-water source is adequate. The failure of a valve, fitting, or pump should not cause more than a momentary loss of water to the

engines operating. It is sometimes difficult to protect against all possible failures in the cooling-water system, but much can be done through proper design to provide a maximum of operating continuity.

The piping arrangement used in any particular installation will be governed to a large extent by the general plant arrangement. Each plant is a special case and must be considered in the light of all influencing features. Typical schematic diagrams for cooling-water piping using a cooling tower and with either a single- or double-circuit system are shown in Fig. 125. The double-circuit cooling system is superseding the single-circuit arrangement in most present-day installations.

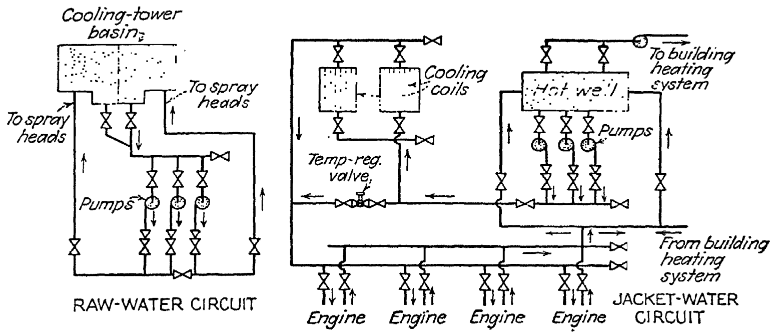


FIG. 126.—Cooling-water-piping arrangement designed for ease of cleaning and repair and continuity of service.

A recent cooling-system design for a plant with four engines installed is shown in Fig. 126. In this layout the designers provided an arrangement that allowed one-half of the cooling system to be taken out of service for maintenance or repairs while maintaining continuity of plant operation. Provisions were also made for future extensions through the addition of cooling facilities or pumps without interfering with the operation of the existing system.

The designs so far considered employed cooling towers as a source of cooling water. If a river or lake is available for supplying the cooling water, then a double-circuit arrangement using shell-and-tube heat exchangers for cooling the jacket water can be employed as shown schematically in Fig. 127.

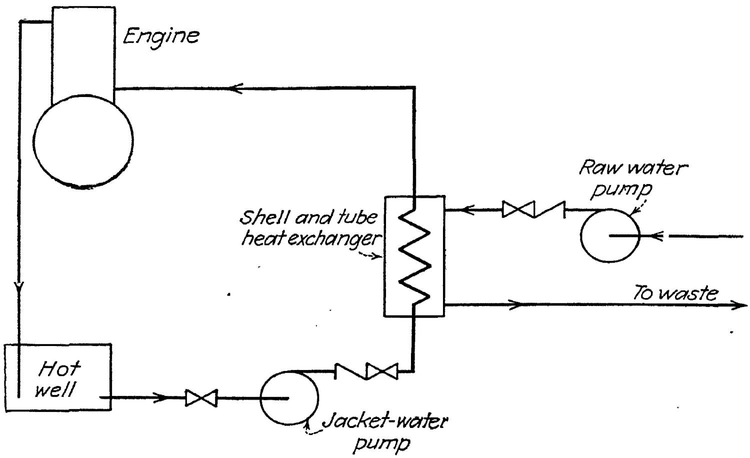


FIG. 127.—Arrangement using shell-and-tube heat exchanger for cooling jacket water.

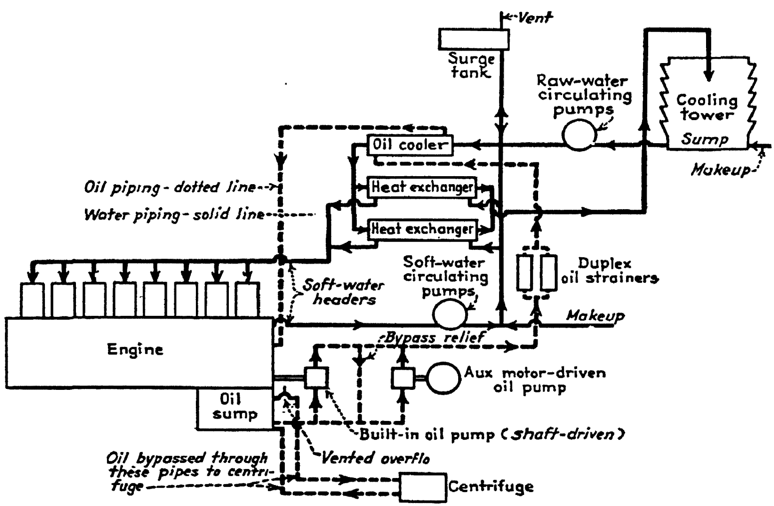


FIG. 128.—Schematic diagram of oil and cooling-water piping for gas-diesel installation. (*Courtesy of Power.*)

In addition to cooling engine jackets, it is necessary in many instances to provide cooling for the lubricating oil. This cooling is universally done by means of a shell-and-tube heat exchanger. Cooling water may be taken either from the usual engine jacket-water supply or from the raw-water side of a double-circuit cooling system. Such an arrangement for lubricating-oil cooling and engine cooling is shown in Fig. 128.

When planning the cooling-water system, provisions must be made for expansion of the water in the engine jacket-water circuit. This can be done either by installing a hot well or by providing an expansion tank equipped with a suitable overflow in the circuit. Whenever an expansion tank is used it must be located above the hydraulic gradient of the line at the point of installation in order that the jacket cooling water will remain in the system and not be drained out through the expansion-tank overflow. When a hot well is used, the discharge line into the hot well should be extended below the water level in the well in order to make effective use of the siphonic action resulting.

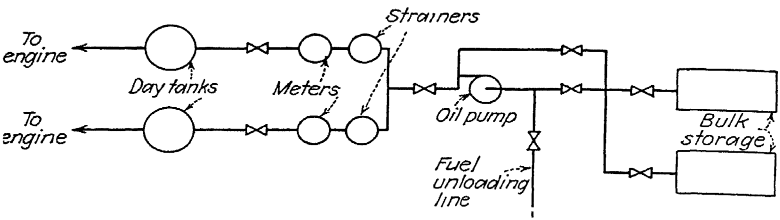
155. Fuel-oil Piping.—The fuel-oil system consists of the pipe, valves, fittings, and other accessories required for delivering fuel oil to each engine as required. In the case of large stations, this involves transfer from railroad tank car to bulk storage, from bulk storage to the day tank for each engine, and from the day tank to the engine as needed. In the small industrial plant, fuel may be delivered to bulk storage by tank wagon. Consequently piping for conveying the oil from railroad tank cars is not required.

Schematic diagrams for two typical fuel-oil piping systems are shown in Fig. 129. In both instances fuel-oil strainers and meters are provided between the main or bulk storage tanks and the engine day fuel tanks.

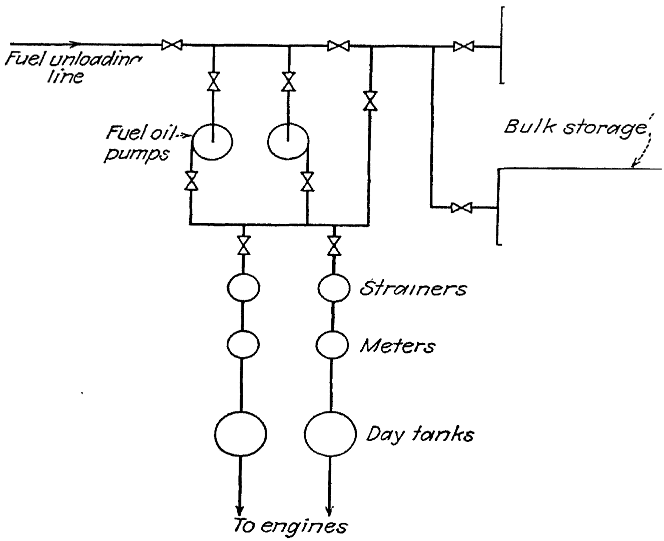
Care must be taken when planning the fuel-oil piping system to see that rules of the National Board of Fire Underwriters as well as individual state requirements are complied with in the design. This applies particularly to capacity and number of day tanks, location of day fuel tanks in the engine room, and provisions for valving of oil lines to eliminate fire hazards.

In northern latitudes during severe winter weather, and particularly in those installations employing high-viscosity fuels, provisions must be made for heating the fuel during

extremely cold weather. This heat is usually supplied from the engine-jacket-water discharge which is piped through coils in the bulk storage tanks. In some cases, it is necessary to provide



(a)-FUEL UNLOADING SYSTEM EMPLOYING SINGLE OIL PUMP



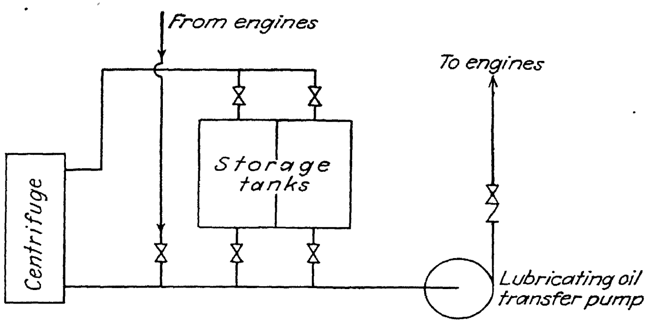
(b)-FUEL SYSTEM EMPLOYING TWO OIL PUMPS

FIG. 129.—Typical fuel-oil handling systems.

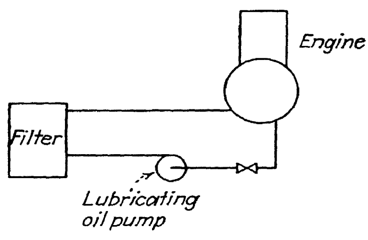
steam for this purpose by means of a waste-heat boiler or an auxiliary-station boiler.

156. Lubricating-oil Piping.—The extent and character of the lubricating-oil piping will be influenced to a large degree by the type of engines installed and the quantity of lubricating oil to

be purified. In the plant having 100 hp installed, a 50-gal barrel may be the extent of the lubricating-oil stock, while in the station having several thousand horsepower installed, the lubricating oil to be handled may range from 10,000 to 25,000 gal. It is apparent, therefore, that the layout of the lubricating-oil piping is governed by the quantity of oil to be handled and the reconditioning necessary for the crankcase oil.



(a)-Lubricating system serving a group of engines



(b)-Lubricating system for a single engine

FIG. 130.—Schematic diagrams of simple lubricating-oil-piping systems.

All lubricating-oil piping forming an integral part of the engine is provided and installed by the engine builder. The piping necessary for removing the lubricant from the engine, reconditioning it, and returning it to the engine must be planned as a coordinated part of the plant. This oil piping, primarily for the purpose of reconditioning the lubricating oil used by the several engines, must be planned in conjunction with the type

of lubricating-oil reconditioning system used. Thus the piping plan for continuously centrifuging the lubricating oil will be different from that utilized for batch reclaiming. There are so many different arrangements possible for lubricating-oil piping that no attempt will be made to depict all of them. Rather, some of the simpler lubricating-oil piping systems now in use are shown in Fig. 130 merely as an indication of what is involved in the planning of an adequate lubricating-oil system.

157. Starting-air Piping.—Most diesel engines, and particularly the large slow-speed units, use compressed air for starting.

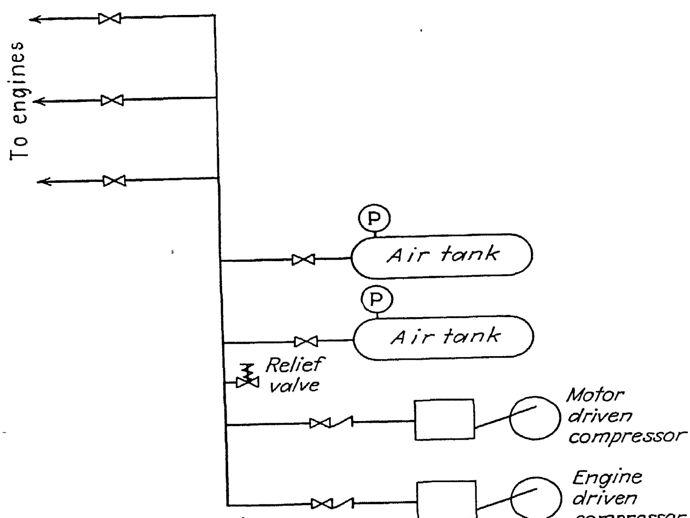


FIG. 131.—Starting air-piping system.

This necessitates the installation of air compressors, air tanks, and piping for connecting these with the engine-starting mechanism. Air pressure used for most starting systems is 250 psi, although some systems employ pressure exceeding this value, and others use a lower pressure.

Tanks, piping, and valves used for this service must be designed to withstand the operating pressures employed. Furthermore, precautions must be taken to ensure that pressures exceeding

this value do not occur or serious damage may result from failure of air tanks or piping.

Compressed-air tanks must be constructed in compliance with the Code for Unfired Pressure Vessels of the A.S.M.E. or applicable local and state codes which may take precedence. Piping must comply with the Code for Pressure Piping of the American Standards Association, or local or state codes applying.

Pipe and fittings are frequently galvanized for protection against internal corrosion caused by the presence of moisture in the air. Suitable drip pockets should be installed in the air system to catch the accumulated moisture. Safety valves of suitable size must be installed in the air system and set to open to a pressure not more than 10 per cent greater than the normal working pressure. Each air receiver should be equipped with a pressure gauge.

A typical layout of the starting-air system for a plant is shown in Fig. 131.

158. Exhaust and Air-intake Piping.—Pipe and fittings for engine exhaust and air intakes are usually purchased with the engine, and therefore furnished by the engine builder. It is usual, however, for the plant designer to assist with the layout of these lines since their location and extent are dependent upon the design of the plant building and the space provided for them.

In general, steel pipe, either seam or spiral weld, is used for the air intake. Where standard-weight pipe is used, it is advantageous and desirable to weld the line throughout.

The exhaust is fabricated with standard-weight welded-steel pipe. If spiral-weld pipe is used, the pulsating exhaust gases set up drumming noises in the pipe. The exhaust line should be welded and provisions made for absorbing the expansion in the line either by the use of a corrugated or slip-type expansion fitting.

Typical exhaust assemblies are shown in Fig. 132. In general, the assembly shown by *A* is preferred since it requires a minimum length of exhaust piping and a single 90-deg. bend in the pipe. The assembly shown in *C* is not satisfactory where a traveling crane is installed, while assembly *B* involves considerable pressure drop in the line between the engine and muffler.

159. Pipe Material.—Pipe can be obtained in a wide variety of materials and for practically every type of service where fluids

or gases must be transported. Pipe is usually considered as being made of steel or cast iron, although copper, brass, glass, aluminum, lead, tile, concrete, wood, transite, fiber, plastics, and many other materials are used for its construction. Practically all pipe used in internal-combustion-engine plants is made

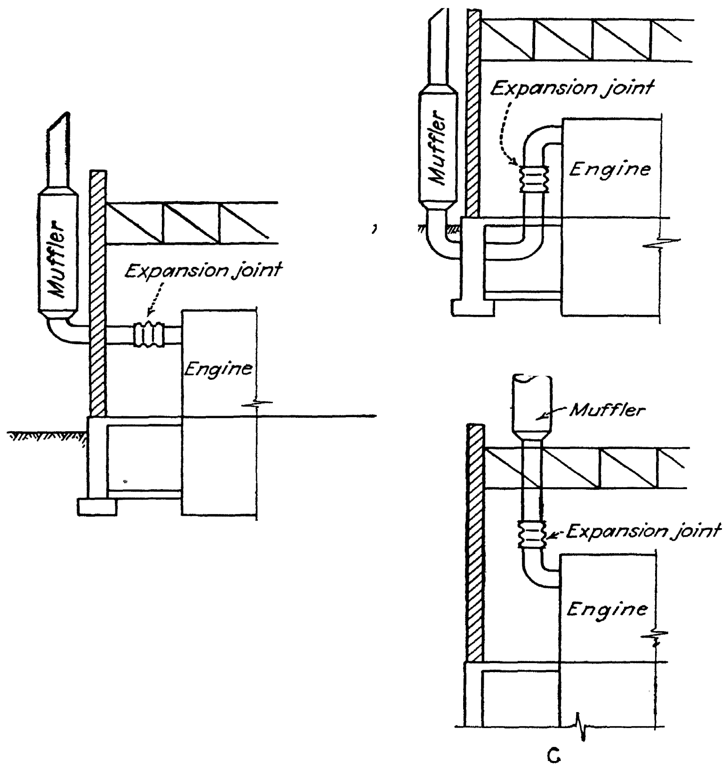


Fig. 132.—Typical exhaust-piping installations.

from steel, cast iron, wrought iron, and copper. Some conditions may require the use of other pipe materials, but the four mentioned constitute by far the bulk of the pipe employed for this service. Those pipe materials normally used in internal-combustion-engine power plants are set forth in Table 39.

Table 39.—Pipe Materials Used in Internal-combustion-engine Power Plants

Service	Pipe material					
	Steel			Wrought iron	Cast iron	Copper
	Seamless	Welded	Spiral welded			
Fuel injection.......						
High-pressure air...		x				
Fuel-oil supply......		x				
Lubricating oil......		x				
Air intake..........		x				
Exhaust...........		x				
Cooling-water lines:						
Inside plant......				x		
Outside plant.....				x		

With the exception of starting-air lines, which carry a pressure of approximately 250 psi, and the fuel-injection and high-pressure air-injection lines, which range in pressure from 600 to 2,000 psi, all the plant piping is under relatively low pressure. It is therefore satisfactory to use standard-weight iron and steel pipe for such services as circulating water, fuel-oil transportation, exhaust and air-intake lines, and lubricating-oil lines. The Code for Pressure Piping permits the use of steel, wrought iron, copper, and brass piping for air pressures above 250 psi, although wrought iron cannot be used for air pressures exceeding 400 psi. Reference should be made to this code, or other codes applicable in the locality where the plant is to be installed, for rulings pertaining to the types of piping materials suitable for high-pressure service.

160. Pipe Fittings.—Pipe fittings suitable for internal-combustion-engine power-plant service are made from cast iron, malleable iron, cast steel, forged steel, bronze, and brass. Copper fittings are usually employed wherever copper pipe is used for lubricating-oil lines on engines, and in these special cases the fittings are usually sweated to the pipe with either hard or soft solder. For most service, cast- and malleable-iron fittings are used since relatively low pressures are encountered and fittings of these materials are cheap.

TABLE 40.—DIMENSIONS OF LAP-WELDED AND SEAMLESS STEEL PIPE[a]
A.S.T.M. Designation A 106-36

Nominal pipe size, in.	OD, in.	Standard pipe and OD sizes 3/8 in. thick	Extra strong pipe and OD sizes 1/2 in. thick	Double extra strong	Nominal wall thickness, in. Schedule number[b]					
					10	20	30	40	60	80
1/8	0.405	0.068	0.095	.....	.....	.....	.....	0.068	.....	0.095
1/4	0.540	0.088	0.119	.....	.....	.....	.....	0.088	.....	0.119
3/8	0.675	0.091	0.126	.....	.....	.....	.....	0.091	.....	0.126
1/2	0.840	0.109	0.147	0.294	.....	.....	.....	0.109	.....	0.147
3/4	1.050	0.113	0.154	0.308	.....	.....	.....	0.113	.....	0.154
1	1.315	0.133	0.179	0.358	.....	.....	.....	0.133	.....	0.179
1 1/4	1.660	0.140	0.191	0.382	.....	.....	.....	0.140	.....	0.191
1 1/2	1.900	0.145	0.200	0.400	.....	.....	.....	0.145	.....	0.200
2	2.375	0.154	0.218	0.436	.....	.....	.....	0.154	.....	0.218
2 1/2	2.875	0.203	0.276	0.552	.....	.....	.....	0.203	.....	0.276
3	3.500	0.216	0.300	0.600	.....	.....	.....	0.216	.....	0.300
3 1/2	4.000	0.226	0.318	0.636	.....	.....	.....	0.226	.....	0.318
4	4.500	0.237	0.337	0.674	.....	.....	.....	0.237	.....	0.337
5	5.563	0.258	0.375	0.750	.....	.....	.....	0.258	.....	0.375
6	6.625	0.280	0.432	0.864	.....	.....	.....	0.280	.....	0.432
8	8.625	0.277	0.500	0.875	.....	0.250	0.277	.....	0.406	0.500
8	8.625	0.322	.....	.....	.....	.....	.....	0.322		
10	10.750	0.307	0.500	.....	.....	0.250	0.277	.....	0.500	0.593
10	10.750	0.365	.....	.....	.....	.....	.....	0.365		
12	12.750	0.330	0.500	.....	.....	0.250	0.277	0.406	0.562	0.687
12	12.750	0.375								
14 OD	14.000	0.375	0.500	.....	0.250	0.312	0.375	0.437	0.593	0.750
16 OD	16.000	0.375	0.500	.....	0.250	0.312	0.375	0.500	0.656	0.843
18 OD	18.000	0.375	0.500	.....	0.250	0.312	0.437	0.562	0.718	0.937
20 OD	20.000	0.375	0.500	.....	0.250	0.375	0.500	0.593	0.812	1.031
24 OD	24.000	0.375	0.500	.....	0.250	0.375	0.562	0.687	0.937	1.218

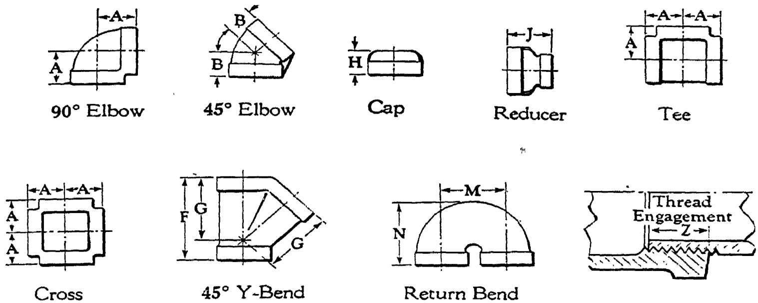
[a] The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions and include an allowance for mill tolerance of 12.5 per cent under the nominal thicknesses.

[b] The schedule numbers indicate approximate values of the expression: $1{,}000 \times \frac{P}{S}$, where P = the internal pressure in pounds per square inch and S = the allowable fiber stress in pounds per square inch.

Connections are usually screwed or flanged. Welded connections are coming to be used more extensively in internal-combustion-engine plants since the number of joints can be reduced materially and assemblies are produced that can be installed in limited quarters more advantageously than assemblies employing either screwed or flanged and bolted fittings. For

TABLE 41.—STANDARD CAST-IRON SCREWED FITTINGS

Working pressures 125 psi steam, 175 psi cold water, oil, or gas, nonshock



Size	*A*	*B*	*F*	*G*	*J*	*H*	*Z*	Return bends: Close pattern Size	Close *M*	Close *N*	Open pattern Size	Open *M*	Open *N*	Wide pattern Size	Wide *M*	Wide *N*
1/4	1 3/16	3/4	...	.....	.....	....	3/8									
3/8	1 5/16	1 3/16	...	.....	.....	....	3/8									
1/2	1 1/8	7/8	...	.....	.....	....	1/2	1/2	1 1/4	1 23/32				1	3	3
3/4	1 5/16	1	3	2 1/4	1 1/2	....	9/16	3/4	1 1/2	2 1/32	3/4	1 7/8	2 7/32	1	4	3 1/2
1	1 1/2	1 1/8	3 1/2	2 3/4	[a]1 7/8	....	11/16	1	1 3/4	2 3/8	1	2 1/2	2 11/16			
1 1/4	1 3/4	1 5/16	4 1/4	3 1/4	2 1/8	....	11/16	1 1/4	2 1/4	2 29/32	1 1/4	3	3 9/32			
1 1/2	1 15/16	1 7/16	4 7/8	3 13/16	2 1/4	....	11/16	1 1/2	2 1/2	3 1/4	1 1/2	3 1/2	3 3/4			
2	2 1/4	1 11/16	5 3/4	4 1/2	2 7/16	....	3/4	2	3 1/4	3 31/32	2	4 1/2	4 19/32	1 1/4	4	3 3/4
2 1/2	2 11/16	1 15/16	6 3/4	5 3/16	2 5/8	....	15/16				2 1/2	5 1/2	5 7/16	1 1/4	6	4 3/4
3	3 1/8	2 3/16	7 7/8	6 1/8	2 7/8	....	1				3	6 1/2	6 5/16			
3 1/2	3 7/16	2 3/8	...	.....	....	....	1 1/16							1 1/2	6	5
4	3 3/4	2 5/8	9 3/4	7 5/8	3 3/8	2 1/16	1 1/8							2	6	5 5/16
5	4 1/2	3 1/16	...	.....	3 7/8	2 3/8	1 1/4									
6	5 1/8	3 7/16	...	.....	4 3/8	2 5/8	1 5/16									
8	6 9/16	4 1/4	...	.....	5 1/4	3 1/8	1 7/16									

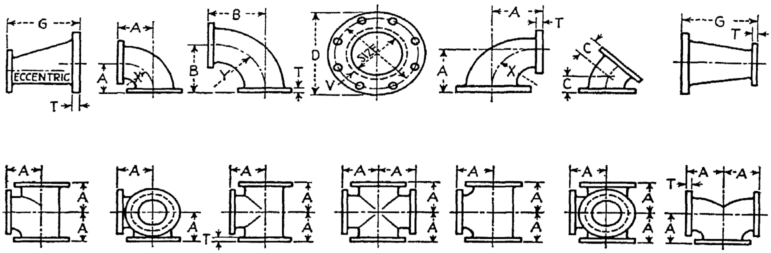
Reprinted courtesy of Crane Co.

[a] 1- by 1/2-in. reducers are 1 11/16 in. end to end.

pipe sizes under 4 in. the screwed connection is probably the simplest and most economical available since it permits field fabrication and assembly of the pipe with a minimum of difficulty.

Considerable attention is being given to the use of compression-type couplings and fittings with plain-end steel pipe. These fittings are designed to provide for expansion, flexibility, and

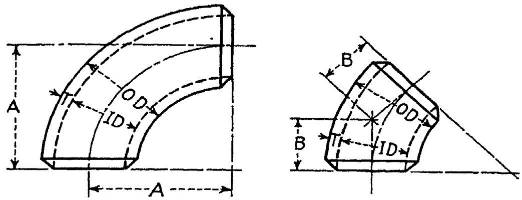
TABLE 42.—AMERICAN STANDARD FLANGED FITTINGS—125 LB
Cast-iron, Straight Sizes



Pipe size	Flange diam.	Center to face			Radii		Metal thick		Reducer F.F.	Drilling		
		Ell tee cross	Long radii ell	45 deg ell	Standard	Long	Body	Flange		[a]Bolt 1/8 in. smaller — Bolt hole		
	D	A	B	C	X	Y	V	T	G	No.	[a]Diam.	Circ.
1	4 1/4	3 1/2	5	1 3/4	.....	....	7/16	7/16	....	4	5/8	3 1/8
1 1/4	4 5/8	3 3/4	5 1/2	2	2 1/4	....	7/16	1/2	....	4	5/8	3 1/2
1 1/2	5	4	6	2 1/4	3	....	7/16	9/16	....	4	5/8	3 7/8
2	6	4 1/2	6 1/2	2 1/2	3 1/2	5 1/4	7/16	5/8	5	4	3/4	4 3/4
2 1/2	7	5	7	3	4	5 3/8	7/16	11/16	5 1/2	4	3/4	5 1/2
3	7 1/2	5 1/2	7 3/4	3	4 1/2	6 1/4	7/16	3/4	6	4	3/4	6
3 1/2	8 1/2	6	8 1/2	3 1/2	5	6 7/8	7/16	13/16	6 1/2	8	3/4	7
4	9	6 1/2	9	4	5 1/2	7 3/8	1/2	15/16	7	8	3/4	7 1/2
5	10	7 1/2	10 1/4	4 1/2	6 1/2	8 1/2	1/2	15/16	8	8	7/8	8 1/2
6	11	8	11 1/2	5	7	9 5/8	9/16	1	9	8	7/8	9 1/2
8	13 1/2	9	14	5 1/2	7 3/4	12	5/8	1 1/8	11	8	7/8	11 3/4
10	16	11	16 1/2	6 1/2	9 1/4	14 1/8	3/4	1 3/16	12	12	1	14 1/4
12	19	12	19	7 1/2	10 1/2	16 1/2	13/16	1 1/4	14	12	1	17
14	21	14	21 1/2	7 1/2	11 1/4	18 7/8	7/8	1 3/8	16	12	1 1/8	18 3/4
16	23 1/2	15	24	8	12 1/2	21 1/4	1	1 7/16	18	16	1 1/8	21 1/4
18	25	16 1/2	26 1/2	8 1/2	13 1/2	23 5/8	1 1/16	1 9/16	19	16	1 1/4	22 3/4
20	27 1/2	18	29	9 1/2	14 3/4	26	1 1/8	1 11/16	20	20	1 1/4	25
24	32	22	34	11	17	30 3/4	1 1/4	1 7/8	24	20	1 3/8	29 1/2
30	38 3/4	25	41 1/2	15	20 1/2	37 3/4	1 7/16	2 1/8	30	28	1 3/8	36
36	46	28	49	18	23	45	1 5/8	2 3/8	36	32	1 5/8	42 3/4
42	53	31	56 1/2	21	26 1/2	52	1 13/16	2 5/8	42	36	1 5/8	49 1/2
48	59 1/2	34	64	24	29 3/4	59 1/4	2	2 3/4	48	44	1 5/8	56

Courtesy National Valve and Manufacturing Co.

TABLE 43.—WELDING FITTINGS



Nominal pipe diameter	90 deg		45 deg		Schedule 40			Schedule 80		
	Long radius *A*	Short radius *A*	*B*	*A*	OD	ID	*T*	OD	ID	*T*
1	1½	1	⅞	1½	1.315	1.049	0.133	1.315	0.957	0.179
1¼	1⅞	1¼	1	1⅞	1.660	1.380	0.140	1.660	1.278	0.191
1½	2¼	1½	1⅛	2¼	1.900	1.610	0.145	1.900	1.500	0.200
2	3	2	1⅜	3	2.375	2.067	0.154	2.375	1.939	0.218
2½	3¾	2½	1¾	3¾	2.875	2.469	0.203	2.875	2.323	0.276
3	4½	3	2	4½	3.500	3.068	0.216	3.500	2.900	0.300
3½	5¼	3½	2¼	5¼	4.000	3.548	0.226	4.000	3.364	0.318
4	6	4	2½	6	4.500	4.026	0.237	4.500	3.826	0.337
5	7½	5	3⅛	7½	5.563	5.047	0.258	5.563	4.813	0.375
6	9	6	3¾	9	6.625	6.065	0.280	6.625	5.761	0.432
8	12	8	5	12	8.625	7.981	0.322	8.625	7.625	0.500
10	15	10	6¼	15	10.750	10.020	0.365	10.750	9.750	0.500[c]
12	18	12	7½	18	12.750	12.000	0.375[a]	12.750	11.750	0.500[a]
14	21	14	8 11/16	21	14.000	13.250	0.375[b]	14.000	13.000	0.500[a]
16	24	15	9 15/16	24	16.000	15.250	0.375[b]	16.000	15.000	0.500[d]
18	27	16½	11 3/16	27	18.000	17.250	0.375[a]	18.000	17.000	0.500[a]

Courtesy Tube-Turns, Inc.
[a] Thickness does not correspond to any schedule number.
[b] Schedule No. 30.
[c] Schedule No. 60.
[d] Schedule No. 40.

variation in alignment of the pipe. They are relatively easy to install and can be dismantled readily for inspection and cleaning of the pipe line if the necessity should arise.

In general, it will be found advantageous in installing circulating water lines inside the building to weld all lines 4 in. and larger and employ screwed fittings on all smaller lines. Outside the building, cast-iron pipe and fittings are favored for this service.

Compressed air and oil lines 2 in. and larger are usually welded with the smaller lines coupled by means of screwed fittings. Welding fittings must be of steel, while malleable iron for screwed fittings is usually specified.

161. Valves.—Valves are used in a piping system to control the flow of the liquid or gas. This control may permit or stop the flow in a line, permit flow in only one direction, or regulate the

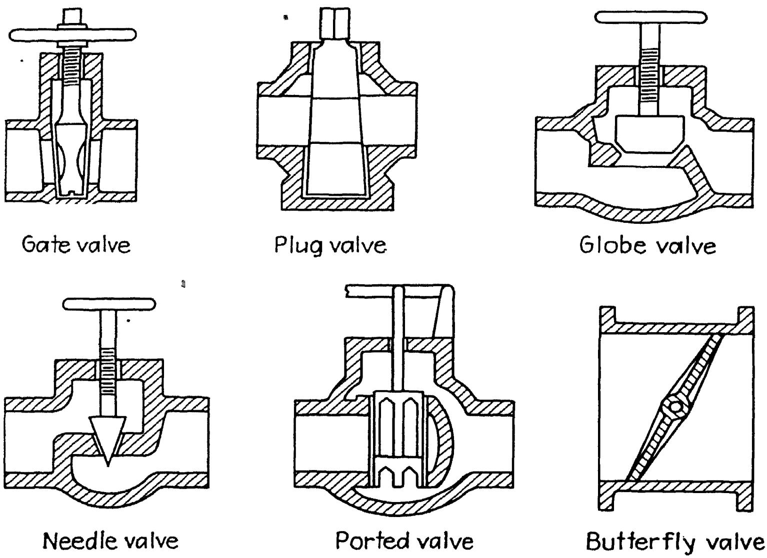


FIG. 133.—Typical valves. (*Courtesy of Power.*)

amount of liquid or gas flowing through the line. These valves may be equipped for manual, electric motor, hydraulic, or full-automatic operation. Automatic valves are generally those regulating the direction or quantity of liquid or gas flowing.

Typical valves are shown in Fig. 133. Of the six types shown the gate, plug, and globe valves are usually employed for opening or closing a line. The needle valve and the globe valve are employed where it is desirable to throttle the flow manually. Ported valves and modifications of the needle valve are used with suitable control accessories to regulate automatically the flow

through the valve. Butterfly valves may be used to regulate flow as well as prevent reverse flow through the line.

Most valves used in internal-combustion-engine power plants will be cast iron or bronze. The only exceptions will be those valves used for high-pressure air and high-temperature service beyond the strength limitations of cast iron and bronze, and in these instances cast- or forged-steel valves must be used.

162. Gate and Globe Valves.—Gate and globe valves are usually used for stopping flow through a line. Globe valves are often employed for throttling service. Gate valves may be employed for throttling although they are subjected to considerable wear when so employed.

Gate valves may be equipped with either solid or split wedges or double disks, although for valves in the sizes used in internal-combustion-engine plants, it is usual to employ the solid wedge gate. Cast-iron gate valves should be provided with suitable seat and disk rings. Bronze gate valves should have bronze rings and should be provided with bronze or stainless-steel stems.

Globe valves should be provided with seat and disk rings as required for gate valves. Union bonnet globe valves with renewable seats and disks are preferred by many.

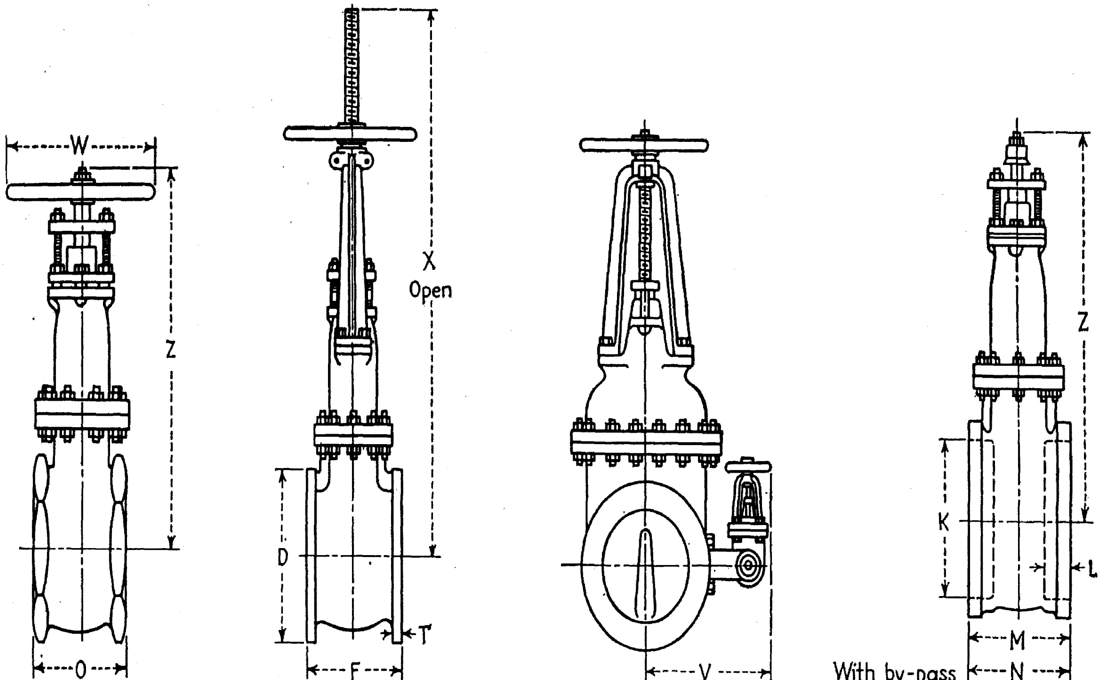
Valves having rising stems are preferred by most operators since the position of the stem gives an indication as to whether the valve is open or closed.

Air and oil lines operating at pressures under 250 psi usually use screwed bronze globe valves for line sizes 1½ in. and less and flanged cast-iron gate valves for line sizes 2 in. and larger. Water lines inside buildings are generally equipped with flanged cast-iron gate valves for line sizes 4 in. and larger, screwed bronze gate valves on 2- and 3-in. lines, and screwed bronze globe valves on lines 1½ in. and smaller. On buried cast-iron water lines, hub-end cast-iron gate valves are used.

In selecting the valves for a plant, it is wise to standardize on one make of valve. Such standardization permits the use of a minimum stock of repair parts and interchangeability of parts between valves of the same size and type.

163. Check Valves.—A check valve is used to prevent reversal of flow in a pipe. Check valves are of many types and design of which those shown in Fig. 134 are the most common. The swing check finds wide application, although it may give rise to unde-

TABLE 44.—STANDARD SCREWED, FLANGED, AND HUB-END GATE VALVES, 125 LB
Inside and Outside Screw and Yoke, Cast-iron Bronze Mounted

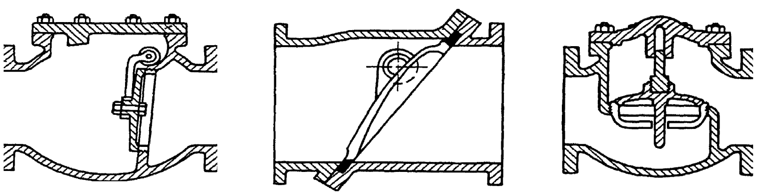


Pipe size	Face-face		Flange		Center to top		By-pass		Wheel diam.	Hub ends				Drilling		
										End to end		Hub		[a]Bolt ⅛ in. smaller		
	Screwed	Flanged	Diam.	Thick-ness	O.S. and Y. open	I.S.	Center to O.S.	Size		Plain	With by-pass	Diam.	Depth	Bolt hole		
	O	F	D	T	X	Z	V		W	M	N	K	L	No.	[a]Diam.	Circ.
2	5 7/16	7	6	5/8	14½	12¼	....	..	6½	8½	..	3¼	2¾	4	¾	4¼
2½	5⅞	7½	7	11/16	16	12¾	....	..	6½	....	..	....	...	4	¾	5½
3	6⅛	8	7½	¾	18½	14¾	....	..	7½	9	..	4¾	2¾	4	¾	6
3½	6½	8½	8½	13/16	20¾	15¼	....	..	7½	....	..	....	...	8	¾	7
4	6⅞	9	9	15/16	23½	16¼	....	..	9	10¼	..	5⅝	3	8	¾	7½
5	7⅜	10	10	15/16	28	19	....	..	10	10¼	..	6¾	3	8	⅞	8½
6	7¾	10½	11	1	31¾	21¼	....	..	12	10¾	..	7⅞	3	8	⅞	9½
8	8¾	11½	13½	1⅛	41	26	....	..	14	12	..	10	3½	8	⅞	11¾
10	9⅞	13	16	1 3/16	49½	31	....	..	16	12¾	..	12¼	3½	12	1	14¼
12	11⅝	14	19	1¼	57½	36	....	..	18	13½	..	14⅜	3½	12	1	17
14	....	15	21	1⅜	66¾	39¼	19½	2	20	13¾	16	16½	3½	12	1⅛	18¾
16	....	16	23½	1 7/16	75¼	44¼	23¾	3	22	16	19	18¾	4	16	1⅛	21¼
18	....	17	25	1 9/16	83½	48¾	24¾	3	24	17	20	20⅞	4	16	1¼	22¾
20	....	18	27½	1 11/16	91¼	52½	27¾	4	24	17½	21	23	4	20	1¼	25
24	....	20	32	1⅞	109	63½	30½	4	30	19	22	27¼	4	20	1⅜	29½
30	....	30	38¾	2⅛	133	75½	34	4	36	30	30	34	4½	28	1⅜	36
36	....	36	46	2⅜	158½	83	39	6	....	....	32	40½	4½	32	1⅝	42¾

Courtesy National Valve and Manufacturing Co.
All dimensions given in inches.

sirable chattering under pulsating or surging flow. The balanced or tilting-disk check valve eliminates much of this chattering and is preferred by many for installation on the discharge side of centrifugal pumps. The lift-type check valve is also used although not so extensively as the other two.

Check valves are used on the discharge of circulating-water pumps, fuel-oil pumps discharging into tanks located above the pump, and in the discharge of air compressors. Check valves should never be installed on the suction side of a pump since the failure of the pump and the closing of the check valve



(a)- Swing check valve (b)-Balanced or tilting disc check valve (c)- Lift-type check valve

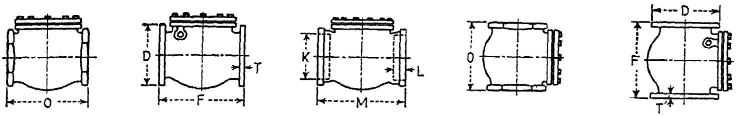
FIG. 134.—Typical check-valve details.

will produce surge pressures which might damage the pump casing.

164. Regulating Valves.—When it is desired to regulate the flow through a line, or to maintain a constant pressure in the line regardless of the rate of flow within the capacity of the regulating valve, a unit similar to that shown in Fig. 135 may be used. The rate of flow is changed by varying the opening of the V-ports in the valve. This adjustment may be effected by a diaphragm arrangement which changes the port openings as the pressure on the discharge side of the valve varies. Control of the valve may also be accomplished by means of temperature changes through suitable control accessories.

In general, valves of this type are used for control of jacket-water flow through heat exchangers to maintain a constant outlet-water temperature from the engine with a constant quantity of water passing through the engine at all loads. They are also used to maintain constant gas pressure delivered to a gas engine regardless of load on the engine or variations in gas pressure above that required.

TABLE 45.—CHECK VALVES—125, 250, AND 800 LB
Cast-iron, Bronze Mounted



Pipe size	Standard 125 lb: Lift type, Horizontal, Screwed E.E. O	Lift type, Horizontal, Flanged F.F. F	Lift type, Vertical, Screwed E.E. O	Swing type, Horizontal or vertical, Screwed E.E. O	Swing type, Horizontal or vertical, Flanged F.F. F	Hub end, Swing type, End to end M	Hub end, Swing type, Diameter K	Hub end, Swing type, Depth L	Flange, Diameter D	Flange, Thickness T	Drilling, Bolt hole, No.	Drilling, Bolt hole, [a]Diam. ([a]Bolt ⅛ in. smaller)	Drilling, Bolt hole, Circ.
2	6½	....	....	6½	8	....	....	...	6	⅝	4	¾	4¾
2½	7	....	7	7	8½	....	....	...	7	11/16	4	¾	5½
3	8	....	8	8	9½	10½	4¾	2¾	7½	¾	4	¾	6
3½	9	....	9	9	10½	....	....	...	8½	13/16	8	¾	7
4	10	11½	10	10	11½	12	5⅝	3	9	15/16	8	¾	7½
5	11¼	13	11¼	11¼	13	13	6¾	3	10	15/16	8	⅞	8½
6	13	14	13	12½	14	14	7⅞	3	11	1	8	⅞	9½
8	....	19½	....	18½	19½	19½	10	3½	13½	1⅛	8	⅞	11¾
10	....	....	....	....	24½	24½	12¼	3½	16	1 3/16	12	1	14¼
12	....	....	....	....	27½	27½	14⅜	3½	19	1¼	12	1	17
14	....	....	....	....	31	31	16½	3½	21	1⅜	12	1⅛	18¾
16	....	....	....	....	36	36	18¾	4	23½	1 7/16	16	1⅛	21¼
18	....	....	....	....	36	32	20⅞	4	25	1 9/16	16	1¼	22¾
20	....	....	....	....	40	35	23	4	27½	1 11/16	20	1¼	25
24	....	....	....	....	48	40	27¼	4	32	1⅞	20	1⅜	29½
30	....	....	....	....	60	50	34	4½	38¾	2⅛	28	1⅜	36

Pipe size	Extra heavy 250 lb: Swing type, Horiz. or Vert., Screwed E.E. O	Swing type, Horiz. or Vert., Flanged F.F. F	Flange, Diam. D	Flange, Thick. T	Drilling, Bolt hole, No.	Drilling, Bolt hole, [a]Diam. ([a]Bolt ⅛ in. smaller)	Drilling, Bolt hole, Circ.	Hydraulic 800 lb (Nonshock): Swing type, Horiz. or vert., Screwed E.E. O	Swing type, Horiz. or vert., Flanged F.F. E	Flange, Diam. D	Flange, Thick. T	Drilling, Bolt hole, No.	Drilling, Bolt hole, [a]Diam. ([a]Bolt ⅛ in. smaller)	Drilling, Bolt hole, Circ.
2	9½	10½	6½	⅞	8	¾	5	10½	12	6½	1¼	8	¾	5
2½	10¾	11½	7½	1	8	⅞	5⅞	12	13	7½	1⅜	8	⅞	5⅞
3	11¾	12½	8¼	1⅛	8	⅞	6⅝	13½	14½	8¼	1½	8	⅞	6⅝
3½	12¼	13¼	9	1 3/16	8	⅞	7¼							
4	13	14	10	1¼	8	⅞	7⅞	15¼	16½	10¾	1⅞	8	1	8½
5	15	15¾	11	1⅜	8	⅞	9¼	17¼	18½	13	2⅛	8	1⅛	10½
6	16½	17½	12½	1 7/16	12	⅞	10⅝	19	20½	14	2¼	12	1⅛	11½
8	....	21	15	1⅝	12	1	13	....	26½	16½	2½	12	1¼	13¾
10	....	24½	17½	1⅞	16	1⅛	15¼	....	30½	20	2⅞	16	1⅜	17
12	....	28	20½	2	16	1¼	17¾	....	34½	22	3	20	1⅜	19¼
14	....	33	23	2⅛	20	1¼	20¼							

Courtesy National Valve and Manufacturing Co.
[a] All dimensions given in inches.

165. Pressure-relief Valves.—Where it is necessary to limit the pressure of a fluid or gas, a pressure-relief or safety valve should be installed. In power plants using internal-combustion engines, safety valves are used on the starting-air system and on the air-injection system of engines so equipped to limit the

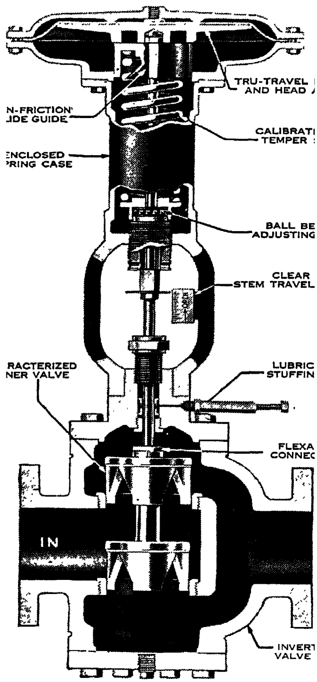


FIG. 135.—Control valve for regulating flow or pressure. (*Courtesy of Fisher Governor Company.*)

pressure to a safe maximum. Capacity data for relief valves are given in Table 46.

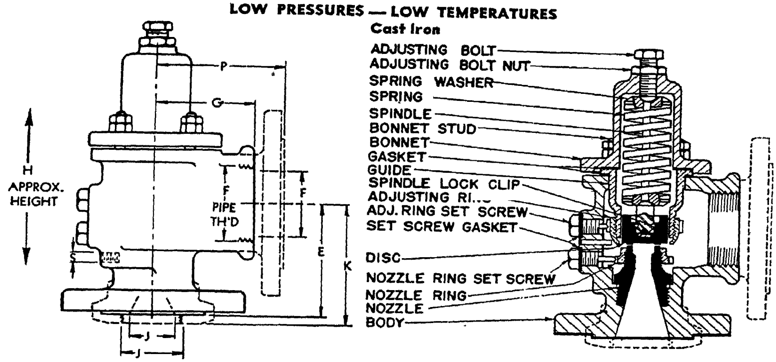
Valves of this type should be checked periodically to make certain they are in good working order.

166. Expansion.—Whenever the temperature of a metallic pipe varies, there occurs a change both in length and diameter of the pipe. The amount of this expansion or contraction is

directly proportional to the dimensions of the pipe and the variation in temperature. In general, only the change in length of the pipe between atmospheric temperature at the time of erection and operating temperature is of interest to the plant designer or pipe erector. When the operating temperatures of the pipe is greater than that at which it is erected, its length increases as

TABLE 46.—CROSBY NOZZLE RELIEF VALVES
Styles *JO* and *JP*

LOW PRESSURES — LOW TEMPERATURES



Pipe thread	Dimensions							
Inlet, in.		1¼				3		
Outlet in.						4		
Orifice						*K*		
Center to face—screwed:								
Inlet		4⅛	5⅛					
Outlet								
American iron flange standard								
Inlet	250	250	250	250	250	250	250	250
Outlet	125	125	125	125	125	125	125	125
Center to face—flanged:								
Inlet								
Outlet						6⅜		9½
Approximate height:								
Plain closed top	12¾	12¾	14⅝				21	28¾
Handwheel	15	15	17	17			23½	33
Screwed cap			14⅞				22	30½
Regular lifting gear						21¾	24	31½
Packed lifting gear		19¼		17½		22	24	36
Drain hole—pipe size	½	½	½		½	½		

TABLE 46.—(*Continued.*)
CAPACITY OF NOZZLE VALVES
Capacity given in cfm at 68 F.

Set pressure, psi gauge	Orifice letter										
	D	E	F	G	H	J	K	L	N	P	Q
10	50	90	140	230	350	580	830	1,290	1,960	2,890	4,980
20	70	125	195	320	500	820	1,180	1,820	2,770	4,080	7,060
30	92	164	256	420	650	1,070	1,520	2,360	3,590	5,280	9,140
40	112	199	312	510	800	1,310	1,870	2,890	4,400	6,480	11,200
50	131	234	369	600	940	1,550	2,210	3,430	5,220	7,670	13,300
60	153	272	427	700	1,090	1,790	2,560	3,960	6,030	8,870	15,400
70	173	308	485	790	1,240	2,040	2,900	4,500	6,850	10,080	17,400
80	195	346	543	890	1,390	2,280	3,250	5,040	7,660	11,300	19,500
90	214	382	600	980	1,530	2,520	3,600	5,570	8,480	12,480	21,600
100	236	420	658	1,080	1,680	2,760	3,940	6,110	9,300	13,700	23,700
120	278	495	773	1,270	1,980	3,250	4,630	7,180	10,900	16,100	27,800
140	320	570	890	1,460	2,270	3,730	5,330	8,250	12,600	18,500	32,000
160	359	640	1,000	1,640	2,570	4,220	6,020	9,320	14,200	20,900	36,100
180	400	715	1,120	1,830	2,860	4,700	6,710	10,400	15,800	23,300	40,300
200	442	790	1,230	2,020	3,160	5,190	7,400	11,500	17,500	25,700	44,400
220	484	862	1,350	2,210	3,450	5,670	8,090	12,500	19,100	28,100	48,600
240	526	935	1,470	2,400	3,750	6,160	8,780	13,600	20,700	30,500	52,800
260	567	1,010	1,580	2,590	4,040	6,640	9,480	14,700	22,300	32,900	56,900
280	610	1,080	1,700	2,780	4,340	7,130	10,200	15,700	24,000	35,300	61,100
300	650	1,160	1,810	2,970	4,630	7,610	10,900	16,800	25,600	37,700	65,200
320	692	1,230	1,930	3,160	4,930	8,100	11,600	17,900	27,230	40,000	69,400
340	735	1,300	2,040	3,350	5,220	8,580	12,200	19,000	28,900	42,400	73,500
360	774	1,380	2,160	3,530	5,520	9,070	12,900	20,000	30,500	44,800	77,700
380	815	1,450	2,270	3,720	5,810	9,550	13,600	21,100	32,100	47,200	81,800
400	856	1,520	2,380	3,910	6,110	10,000	14,300	22,200	33,800	49,600	86,000
420	898	1,600	2,500	4,100	6,400	10,500	15,000	23,200	35,400	52,000	90,100
440	940	1,670	2,620	4,290	6,700	11,000	15,700	24,320	37,000	54,400	94,300
460	980	1,750	2,730	4,480	6,990	11,500	16,400	25,380	38,600	56,800	98,400
480	1,020	1,820	2,850	4,670	7,290	12,000	17,100	26,500	40,300	59,200	102,600
500	1,065	1,900	2,960	4,860	7,580	12,500	17,800	27,500	41,900	61,600	106,700
600	1,270	2,260	3,540	5,800	9,060	14,900	21,200	32,900	50,100	73,600	
700	1,480	2,630	4,120	6,750	10,500	17,300	24,700	38,200	58,200	85,600	
800	1,685	3,000	4,700	7,690	12,000	19,700	28,200	43,600	66,400	97,600	
900	1,890	3,370	5,270	8,640	13,500	22,200	31,600	49,100	74,500	109,600	
1,000	2,100	3,740	5,850	9,580	15,000	24,600	35,000	54,400	82,700		
1,100	2,300	4,110	6,430	10,520	16,500	27,020	38,500				
1,200	2,510	4,470	7,000	11,500	18,000	29,440	42,000				
1,300	2,720	4,840	7,580	12,400	19,450	31,860	45,400				
1,400	2,920	5,210	8,160	13,400	20,900	34,280	48,900				
1,500	3,130	5,580	8,740	14,300	22,300	36,700	52,500				

Courtesy Crosby Steam Gage & Valve Company.

in the case of the engine exhaust, while operating temperatures lower than the erection temperature produce a decrease in the pipe length. Variations in pipe diameter with changes in temperature are usually so small that they may be neglected.

The change in length of a pipe due to temperature variation can be calculated from the equation[1]

$$L_t = L_o\left[1 + a\left(\frac{t - 32}{1{,}000}\right) + b\left(\frac{t - 32}{1{,}000}\right)^2\right] \tag{37}$$

where L_t = length of pipe at temperature t.
L_o = length of pipe at 32 F.
t = final temperature, deg F.
a and b are constants.

The values of the constants a and b are as follows:

Metal		
Cast iron....	0.005441	0.001747
Steel........	0.006212	0.001623
Wrought iron	0.006503	0.001622
Copper......	0.009278	0.001244

Changes in length per 100 ft of pipe for cast iron, steel, wrought iron, and copper at temperatures above 70 F are shown in Fig. 136.

167. Expansion Joints.—Expansion joints may be either the corrugated or slip type, Fig. 137. The former includes all corrugated or bellows expansion joints made of rubber, copper, stainless steel, nickel-steel, and other metals. Metal expansion joints must be used for high-temperature service. Slip joints are used extensively for water, steam, air, gas, and oil lines for pressures below 125 psi and occasionally for pressures up to or even exceeding 250 psi. Joints of the slip type are made in a wide variety of styles to meet the requirements for practically any application within their pressure and temperature limitations. Since slip-type joints are subject to some leakage, they should be placed in locations where maintenance on them can be done conveniently.

[1] WALKER, J. H., and SABIN CROCKER, "Piping Handbook," 3d ed., p. 593, McGraw-Hill Book Company, Inc., New York, 1939.

Whenever an expansion joint is used, the pipe line must be anchored at some point on each side of the joint in order for it

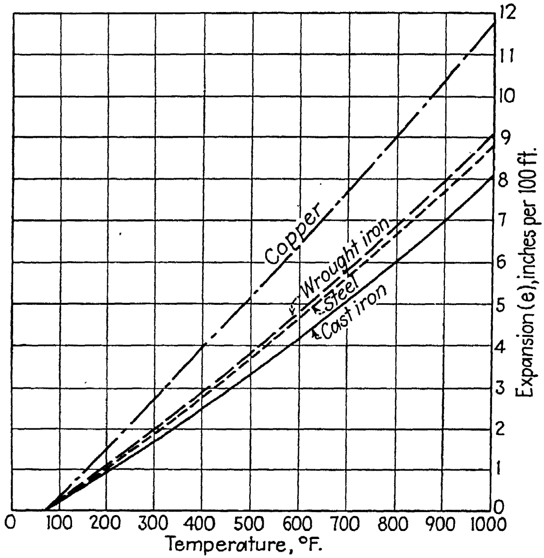


Fig. 136.—Curve showing expansion of pipe materials for temperatures above 70 F.

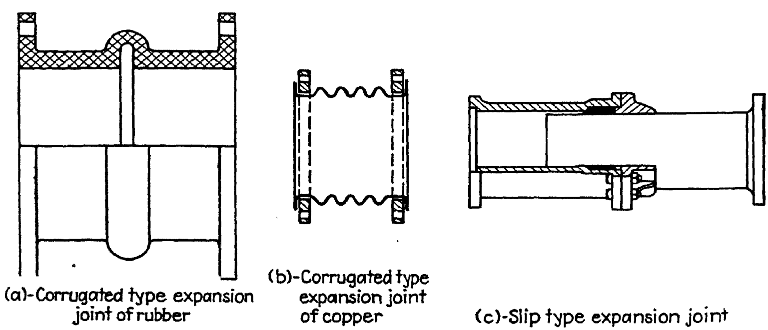


Fig. 137.—Expansion-joint construction details.

to absorb the expansion in the line between anchor points. This expansion force which must be absorbed by each anchor is equal

to the cross-sectional area of the pipe in square inches multiplied by the pressure (pounds per square inch) in the pipe.

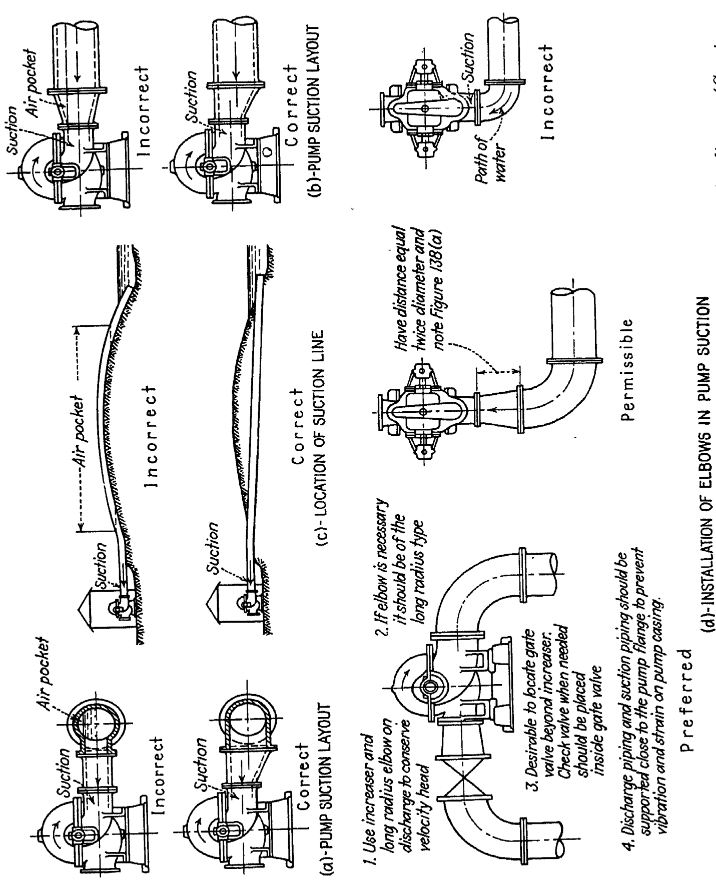


FIG. 138.—Details showing correct and incorrect methods for installing pump suction lines. (*Courtesy of The Hydraulic Institute.*)

168. Installing Centrifugal Pumps.—Centrifugal pumps will give satisfactory, trouble-free service when proper precautions are taken to see that they are installed correctly. In general, the greatest difficulty is experienced with the pump suction,

and consequently considerable attention should be paid to the design and installation of the suction line to each centrifugal pump. Air pockets must be eliminated from the suction to ensure continuous pump operation. Furthermore, the discharge from the pump must be proportioned to conserve the velocity head created at the pump discharge. Figure 138 shows both correct and incorrect methods for installing suction lines connecting to centrifugal pumps.

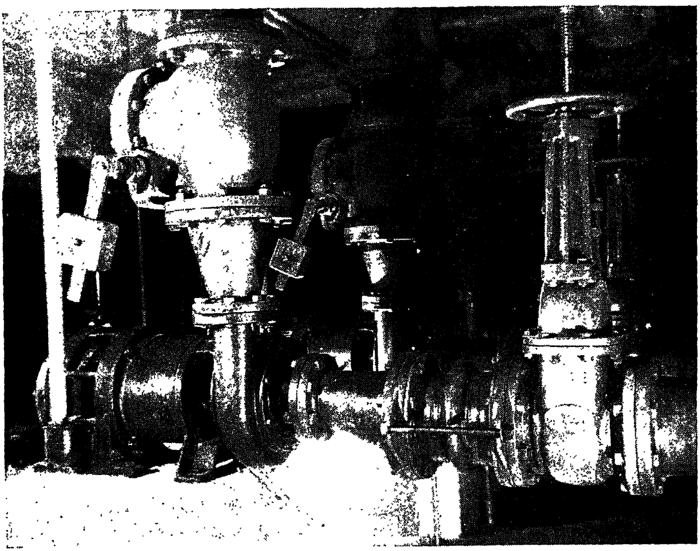


FIG. 139.—Rubber expansion joint installed in pump suction line to facilitate inspection of pump. (*Courtesy of Burns & McDonnell Engineering Company.*)

169. Piping Kinks.—The layout of any piping system involves the exercise of much ingenuity on the part of the designer. Many unique and useful accessories have been produced to fit special conditions.

When installing an end-suction pump, it is the general tendency to connect the pump solidly into the piping system. In order to inspect the pump impeller, it is generally necessary to unbolt the suction and discharge connections, disconnect the electrical

connections to the motor, and remove the pump and motor assembly from its foundation. This difficult task can be eliminated by installing an expansion joint in the pump suction which is equipped with special end plates and take-up bolts, Fig. 139. By tightening up on the bolts the expansion joint can be shrunk and the expansion joint with the reducer connected to the pump suction can be removed to permit quick access to the pump impeller.

The judicious use of unions in a line equipped with screwed connections permits rapid breakdown of the line for maintenance, repairs, or for relocation. Similarly, in welded pipe assemblies it is often advisable to plan the piping so that the valves or other suitable flange joints may facilitate the breakdown of the line as required.

Simplicity should govern the layout. Provisions should be made in the original installation, through the use of suitable valves, for future additions where required without interfering with the operation of the remainder of the piping system. This often requires that the designer give considerable attention to sectionalizing of pipe headers in order to make additions conveniently.

Before putting any piping system into operation, all lines should be thoroughly cleaned out. Considerable care should be exercised to ensure that all dirt, scale, metal chips, rags, wooden plugs, and other materials that would stop flow through a line or prevent the closing of a valve be removed before the line goes into service.

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CHAPTER XV

WASTE-HEAT RECOVERY

The present-day diesel engine, producing a brake horsepower with 7,000 to 8,000 Btu of heat in the fuel, is one of the most efficient prime movers with which we deal. Since the heat equivalent of a horsepower-hour produced is only about one-third of the heat supplied to the engine in the form of fuel, and since engineers are constantly striving to increase the efficiency and economy of equipment with which they deal, it is not strange that much thought has been given to means for utilizing a portion of the heat in the other two-thirds of the fuel.

170. Use of Waste Heat.—In studying the Reports on Oil-engine Power Cost of the A.S.M.E., it is surprising to see what a relatively small percentage of the plants covered by that analysis use waste heat for any purpose. Approximately 140 out of 150 plants covered in the 1938 report dealt with the question of

TABLE 47.—SUMMARY OF WASTE-HEAT RECOVERY DATA

Use of waste heat	Number of plants using waste heat from	
	Exhaust	Jacket water
To heat building	32	20
Heat for slaking lime	0	1
Heating fuel oil	5	8
Heating inlet circulating water	0	2
Heating lubricating oil	3	1
Heating boiler feed make-up	2	1
Hot-water supply	2	2
Heat for process	1	0
Heat not utilized	100	109
Number of services reported	145	144
Number of plants reporting	140	142

NOTE.—The difference between the number of services and the number of plants is due to several plants using waste heat for more than one type of service.

waste-heat recovery. Of those reporting, 72 per cent made no use of heat in the engine exhaust, while 77 per cent made no use of the heat in the jacket water. The majority recovering waste heat used it for building heating only. The analysis of this waste-heat use for various purposes is set forth in Table 47.

For purposes of this chapter the term *waste heat* includes all the heat in the fuel supplied to the engine and not converted into mechanical energy.

Waste-heat utilization to be successful requires a careful study of the characteristics of the machine producing the heat as well as the means for reclaiming it. As a consequence, this chapter considers briefly the characteristics of the waste heat available from the engine as well as the methods that are available for its utilization.

171. The Engine as a Fuel-burning Machine.—The internal-combustion engine is primarily a machine for converting the heat energy of a liquid or gas fuel into mechanical energy. In the conversion process, only about one-third of the available energy in the fuel appears in mechanical form, while the remainder appears in the exhaust, cooling water, and radiation. The utilization of any or all of the heat in the fuel not used for producing mechanical work offers many possibilities for increasing the over-all fuel efficiency.

In order that a portion of this heat normally wasted while operating the engine may be utilized, it is necessary to know its magnitude and characteristics. These qualities are established by means of the following:

1. Heat balance for the particular engine under consideration.
2. Fuel consumption guarantees at full and fractional loads.
3. Cooling-water temperatures on and off the engine at full and fractional loads.
4. Exhaust gas quantities at full and fractional loads.
5. Exhaust gas temperatures at full and fractional loads.

The characteristics of these various items will be considered in the order given.

172. Engine Heat Balance.—Consider a present-day mechanical-injection diesel engine having a full-load rating of 125 hp per cylinder. The head supplied to this engine in the form of fuel is accounted for as follows:

TABLE 48.—HEAT BALANCE FOR DIESEL ENGINE

Engine load	1/4	1/2	3/4	4/4
Useful work, per cent	26	32	33.5	34.5
Cooling water, per cent	24	27	29	30
Exhaust and radiation, per cent	26	26	26	27
Mechanical losses, per cent	24	15	11.5	8.5
Total fuel energy, per cent	100	100	100	100

At full load, 34.5 per cent of the heat energy in the fuel is converted into useful work. Such portion of the remaining 64.5

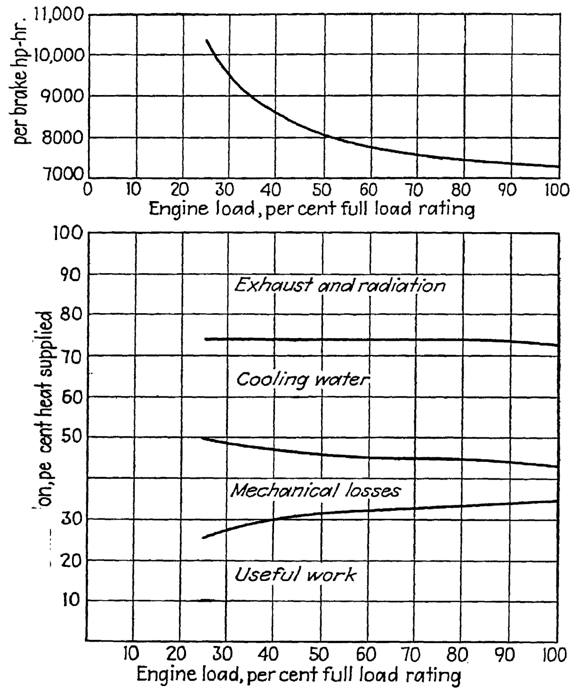


FIG. 140.—Heat balance for 125 hp per cylinder mechanical-injection diesel engine.

per cent as can be put to useful service represents a distinct saving and results in improving the over-all heat utilization.

The heat balance contained in Table 48 is reproduced in graphical form in Fig. 140. While the percentage distribution of the heat in the fuel will vary somewhat from these data for various makes, types, and horsepower ratings of engines, these data do represent fairly average results. The area between the curve of useful work and the line for 100 per cent of heat supplied represents waste heat. The portion shown as "Mechanical losses" cannot be utilized except as it appears as sensible heat for warming the engine room. For waste heat required elsewhere than in the engine room, that in the "Cooling water" and "Engine exhaust" are the only sources. At full load they represent 57 per cent of the heat supplied to the engine in the example given, and this percentage is fairly constant from one-half to full load.

173. Fuel Consumption.—Fuel-oil and natural-gas consumption guarantees for several makes, types, and sizes of modern internal-combustion engines were discussed in Chap. III, Arts. 16 and 17. The variation in heat input to the engine per brake horsepower-hour is shown in Table 49 and is derived from Figs. 14 and 15.

TABLE 49.—VARIATION IN HEAT INPUT TO DIESEL AND GAS ENGINES

Engine load	¼	½	¾	4/4
Diesel engine:				
Btu per bhp-hr max	11,790	9,500	8,550	8,160
Btu per bhp-hr min	9,500	7,600	7,030	7,030
Gas engines:[a]				
Less than 7-in. bore average	21,000	14,500	11,200	10,200
Over 7-in. bore average	18,500	13,000	10,500	9,600

[a] Does not include gas engines operating on the diesel cycle.

174. Cooling-water Temperature.—Cooling-water temperatures have been discussed previously in Chap. XII, Arts. 117 to 120. If the heat transferred from the engine to the cooling water in slow-speed engines is to be utilized at the temperature at which it leaves the engine, then it must be used for purposes where heat at these relatively low temperatures will suffice. Should the temperature of the jacket water off the engine be insufficient, it can be increased by using a portion of the heat in the engine exhaust.

Where high temperature or vapor-phase cooling is employed, the range of uses for the heat in the jacket water is very materially broadened.

175. Exhaust Gas Available.—The quantity of exhaust gas available from a diesel engine is determined by the quantity of air passing through the engine for combustion and scavenging purposes. This weight of air while set by the engine design will vary, depending upon the inlet air temperature and barometric pressure. Data covering the air requirements of various types of engines are given in Table 35.

The theoretical weight of air for combustion per pound of fuel oil is given in Table 50 covering fuel oils from the major oil-producing fields.[1]

TABLE 50.—THEORETICAL QUANTITIES OF AIR FOR OIL COMBUSTION

Source	Specific gravity	Btu per lb oil	Lb air per lb oil, theoretical
Texas........	0.907	19,230	14.15
Mid-continent	0.892	19,376	14.00
California....	0.971	18,820	14.08
Mexico.......	0.986	18,720	13.32

It is readily apparent from a comparison of Tables 35 and 50 that the quantity of air required by a diesel engine per pound of fuel oil consumed is considerably in excess of the theoretical amount required for combustion of the fuel. It is also apparent from Table 35 that a two-stroke-cycle engine requires about twice the amount of air that a four-stroke-cycle engine needs.

176. Exhaust-gas Temperature.—In addition to knowing the quantities of gases discharged through the exhaust in a given time, it is necessary to know the variation in exhaust-gas temperatures at various engine loads. Figure 141 shows the range of exhaust temperatures for two-stroke-cycle mechanical-injection diesel engines, while Fig. 142 gives the same information for four-stroke-cycle mechanical-injection nonsupercharged engines. Exhaust temperatures for air-injection engines, both

[1] DAVIS, RALPH E., HARRY K. IHRIG, DEWEY J. SABIN, and LYON F. TERRY, Economic Utilization of Natural Gas, *Trans., A.I.M. & M.E. Coal Div.*, p. 388, 1931.

two-stroke and four-stroke cycle, will range somewhat lower than those shown in Figs. 141 and 142 owing to the fact that more air per pound of fuel is, in general, required for air-injection than for mechanical-injection engines. Supercharging of four-stroke-cycle engines may tend to reduce the exhaust temperature slightly. Gas engines, operating as they do with a constant air-fuel ratio, have considerably higher exhaust temperatures than do diesel engines. In fact, the difference in exhaust temperature

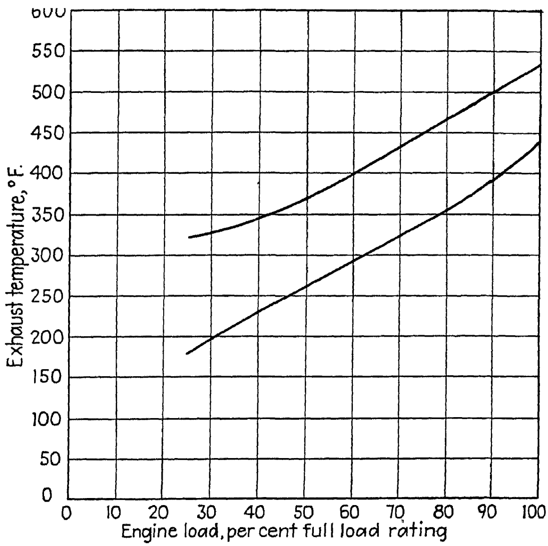


FIG. 141.—Range of exhaust-gas temperatures in mechanical-injection two-stroke-cycle, diesel engines.

between one-fourth and full load is relatively slight as compared with the temperature variation in the diesel exhaust.

These figures indicate higher exhaust temperatures from four-stroke-cycle engines because of the fact that, in general, four-stroke-cycle nonsupercharged engines use less excess air for scavenging and as a result have higher exhaust temperatures.

Where steam is being produced from the heat in a diesel-engine exhaust, the four-stroke-cycle nonsupercharged engine shows up to a better advantage owing to the higher exhaust temperature.

The exhaust from a gas engine, because of its relatively constant temperature, is superior as a steam-producing source to either a two-stroke or four-stroke-cycle diesel or gas-diesel engine.

177. Methods for Utilizing Waste Heat.—This chapter so far has dealt primarily with the characteristics of the two-thirds of the heat of the fuel which does not show up at the engine crankshaft. Let us now consider methods for utilizing a portion of this heat normally rejected by the engine.

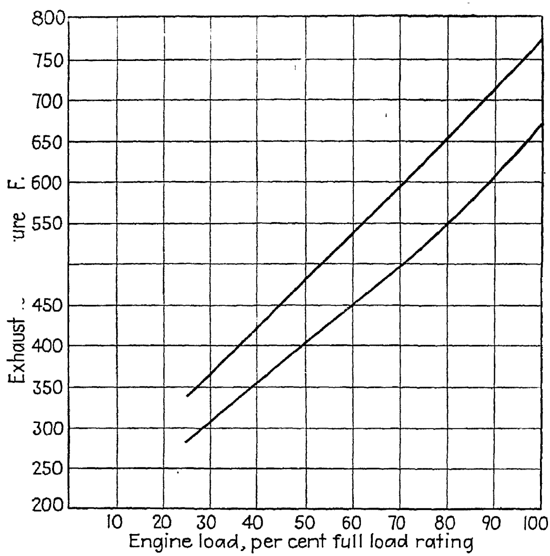


FIG. 142.—Range of exhaust-gas temperatures in four-stroke-cycle, mechanical-injection diesel engines.

It has been claimed, and perhaps rightly, that as much as 85 per cent of the heat in the fuel supplied to a diesel engine can be utilized for crankshaft energy and for heat. The amount of heat reclaimed over and above that appearing as crankshaft energy will depend upon the economic justification of the investment necessary in heat-reclaiming equipment. As previously indicated, the two main sources of waste heat are the jacket-cooling water and the exhaust. Several methods are available for obtaining heat from these sources.

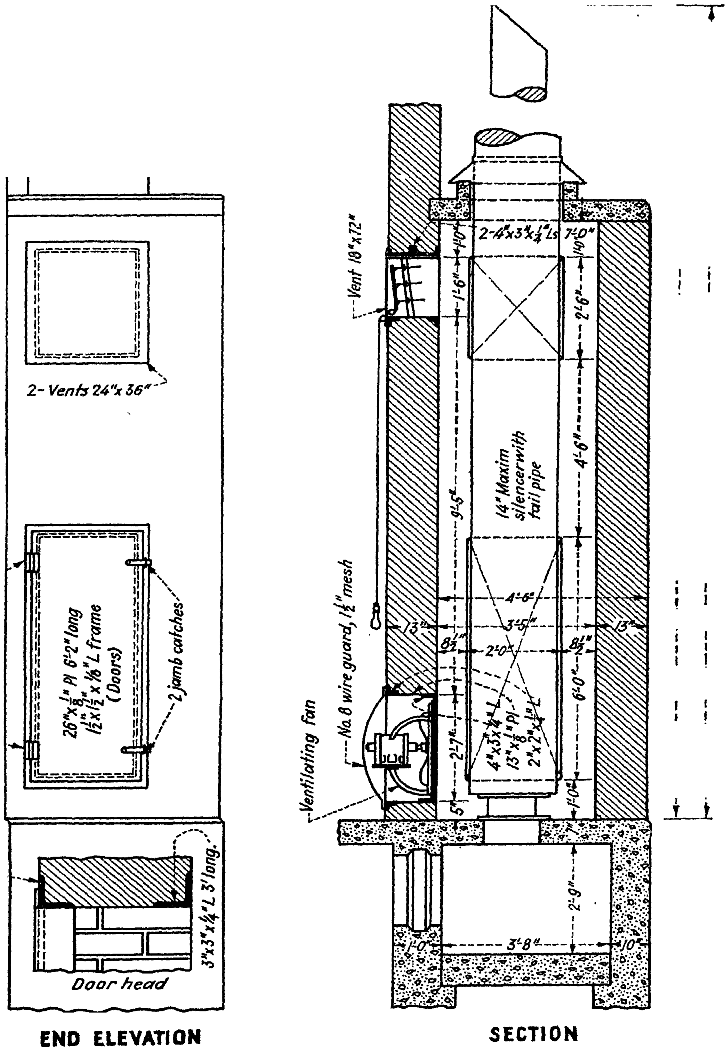


FIG. 143.—Simple waste-heat recovery system for building heating. (*Courtesy of Power.*)

The various methods of heat reclamation fall roughly into two classes, namely, those which do not employ a waste-heat boiler and those that do.

178. Hot-air Heating.—Building heating through the use of hot air has been successfully accomplished by enclosing the engine exhaust silencers and forcing air through the enclosure by means of a fan. One method for accomplishing this is shown in Fig. 143. While no exact formulas are available for calculating the probable heat recovery from such a system, tests at Bloomington, Ill.,[1] show that with engine-exhaust temperatures of 320 F, inlet heating air temperature of 71 F, and outlet temperature of 115 F, a heat recovery of 138,000 Btu per hr was realized when circulating 3,000 cfm of heating air around the silencer. During this test, the engine was operating at approximately half load of 320 hp. From these data it appears that heating systems of this character will serve satisfactorily and furnish sufficient heat for a power-plant building when 5 cu ft of air per minute per rated engine horsepower are circulated around the silencer. This rule-of-thumb guide in designing a heating system may not prove successful in all cases. The designing of a satisfactory heating system such as shown in Fig. 143 requires that a sufficient quantity of air pass over the outside of the silencer in a given time to absorb up to a maximum of 30 per cent of the heat in the exhaust gas. Putting this into the form of an equation, we obtain

$$0.30H = KV(120 - t) \tag{38}$$

where H = total heat in Btu passing through the exhaust silencer per minute.

V = cubic feet of heating air circulated around silencer per minute at the entering air temperature.

t = inlet temperature of heating air.

Values of K vary with inlet air temperature as follows:

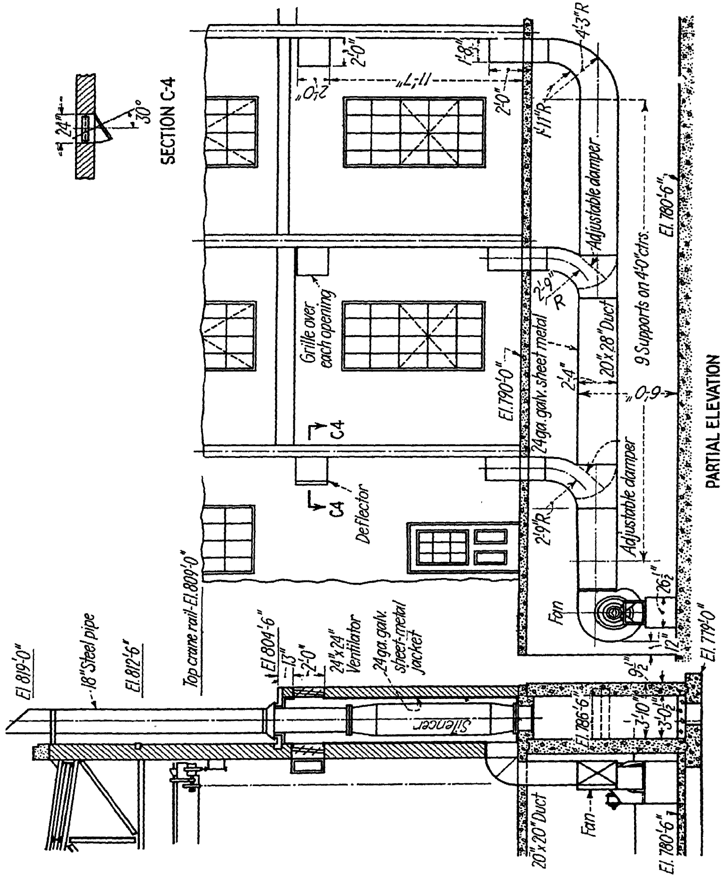
Entering air temp., deg F.	55	60	65	70	75
K....................	0.01851	0.01834	0.01816	0.01799	0.01782

This equation is based upon the assumption that the temperature of the hot air leaving the muffler would be 120 F, with the

[1] HOWELL, LLOYD, Recovering Diesel Waste Heat at Bloomington, *Power*, vol. 77, No. 7, p. 285, June, 1933.

heat carried per cubic foot of dry air determined by the density of the incoming air.

In installations where several engines are employed, an arrangement for building heating shown in Fig. 144 has been used



successfully. This scheme employs a single motor-driven blower and connecting duct work from the blower to all silencers. Dampers in the duct work allow the air circulation to be directed

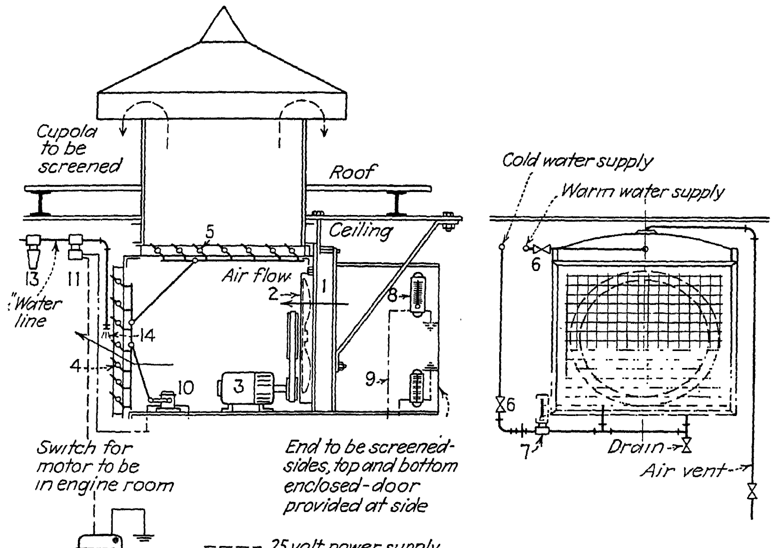
to the active silencers, or to be divided between the silencers in such a manner as to secure the most effective heat distribution in the building. An installation of this type is now in operation in a power plant with three 650-hp diesel engines. The fan provides 4,500 cu ft of air per minute at a static pressure of 1 in. of water. The entire cost of fan, duct work, brick enclosures around the silencers, and adjustable louvers was approximately $750, a low cost for the heating system in a building approximately 40 by 70 ft.

Commercially designed air heaters are also used for reclaiming heat from the exhaust gases. Data available on these units indicate that as much as 35 per cent of the available heat in the engine exhaust at full load can be reclaimed by their use.

179. Hot Water from Exhaust.—Hot water may be obtained from the heat in the exhaust through the installation of a pipe coil in the engine muffler, or a water jacket around the exhaust line. A pipe coil consisting of 2 ft of 1¼-in. pipe per rated engine horsepower when operating with a two-stroke-cycle engine will recover approximately 400 Btu per bhp-hr output at full load, or about 27 per cent of the heat in the exhaust. This is on the basis of clean tubes. When such coils are installed in exhaust lines or mufflers, provisions should be made to facilitate removal for cleaning. A hot-water jacket constructed around the engine exhaust line will absorb roughly 3,000 Btu per hr per sq ft of pipe in contact with the water. This figure is based upon an engine operating at full load, exhaust temperatures ranging between 400 and 450 F, and a heating area of 0.1 sq ft per rated engine horsepower. At fractional loads on the engine the heat transfer to water from the exhaust gas will be less because of the lower exhaust temperatures. This applies both to the pipe coil and the water jacket.

180. Heat from Cooling Water.—Building heating can be accomplished by using the heat in the cooling water as is the case in a manufacturing plant employing diesel engines for its power supply where heating facilities for the building and the cooling-water system for the engines are combined. The cooling water is circulated through six unit heaters located in various parts of the plant. Each unit heater, Fig. 145, is equipped with a fan for forced air circulation and two sets of adjustable louvers thermostatically operated. One set of louvers is on

the face opposite the fan, and when open the circulated air is forced out into the room. The other set of louvers is in the



PART NO.	NO. REQ.	DESCRIPTION
		For hand control
1	1	Radiator
2	1	Fan
3	1	Motor
4	1	Supply damper
5	1	Exhaust damper
6	2	Shut-off valve
7	1	Thermometer
		For automatic control
		Add:
8	1	Room type thermostat
9	1	fyDh 183 Hygrostat
10	1	eyCc Damper controller
11	1	eyBs 712 Solenoid valve
12	1	yBa 270 Power box
13	1	Strainer
14	1	Spray nozzle

FIG. 145.—Jacket-water heat used for automatic heating, ventilating, and humidification. (*Courtesy of Fairbanks, Morse & Company.*)

top of the unit heater, and when open circulating air is forced out through a ventilator in the roof.

When the temperature inside the building falls below 70 F, a thermostat closes the louvers in the top of the unit heater and opens the set admitting the warm air into the building. As soon as the room temperature reaches 70 F, the thermostat closes the louvers admitting air into the room and opens the louvers in the top of the heater, thereby forcing the warm air out of the building.

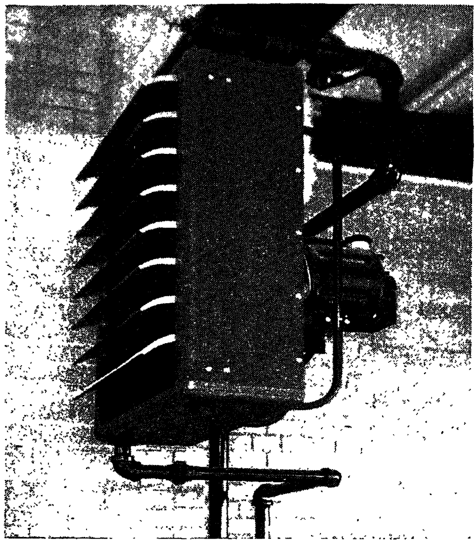


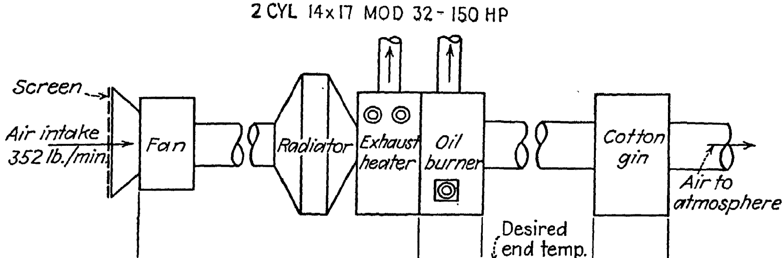
FIG. 146.—Unit heaters utilizing hot water from engine-cooling-water discharge for building heating. (*Courtesy of Burns & McDonnell Engineering Company.*)

This arrangement has the following features:

1. Eliminates a cooling tower for conditioning jacket water.
2. Provides a closed cooling system for the diesel engines installed.
3. Provides heat for the plant throughout the heating season.
4. Provides ventilation for the plant during the remainder of the year.

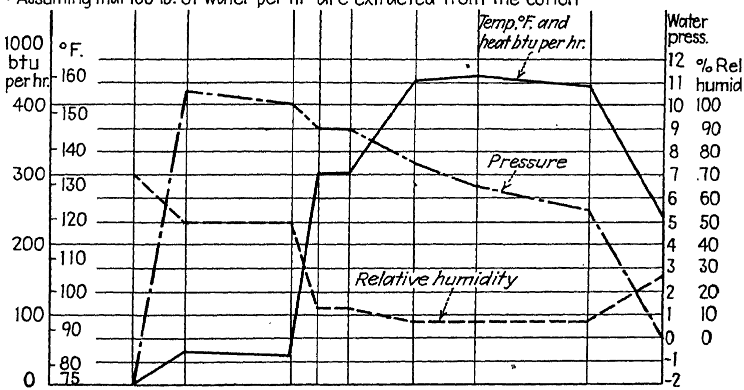
Since the diesel engines operate only during the day, two small oil-fired heating boilers are installed for maintaining room temperatures at night during the heating season.

A heating coil, installed in the exhaust pit, common to the engines in the plant, provides sufficient hot water for washroom purposes. This is the only use made of the heat in the exhaust.



Temp. °F.	75		84		83		132		158		160		158		120
Press. in wat.	0		10.5		10		9		7.5		6.5		5.5		0
Rel. hum. %	70		51		52		13		7		7		7.5		25 lb.*
Heat, btu per hr. Added		46,575		—		256,000		131,950		10,350		—		—	—
Subtr.		—		5,175		—		—		—		10,350		196,650	—
Total		46,575		41,400		297,400		429,350		439,700		429,350		232,700	232,700

*Assuming that 100 lb. of water per hr are extracted from the cotton



Air temperature, pressure and humidity-heat content of air blown through radiator, fan, exhaust heater, oil burner and cotton gin for drying of cotton at 100% engine load

Fan hp. = 20.0 hp.

FIG. 147.—Arrangement for using engine waste heat in a cotton gin. (*Courtesy of Fairbanks, Morse & Company.*)

The heat contained in the jacket water of an engine can be employed for building by the use of conventional unit heaters,

Fig. 146. The unit heater shown is designed to dissipate 97,200 Btu per hr with the hot water entering the heater at 140 F and the air entering at 60 F. Water flow through the heater is 14.6 gpm.

181. Use of Waste Heat in Industry.—Many industrial establishments using internal-combustion engines for producing power are also recovering heat from the jacket water and the exhaust for process applications. Figure 147 gives an application of waste-heat recovery to a cotton gin. Air is needed for transporting the cotton in the ginning process, and heat is required for drying the cotton. That these requirements have been fully met is apparent from a study of Fig. 147.

Heat recovery from engines in dairies, ice plants, and other industrial establishments is being used to an increasing extent and will continue to be used wherever economically feasible.

182. Waste-heat Boilers.—The heat-reclaiming methods so far considered have not required any changes in the normal arrangement of engine auxiliaries. There remains for consideration one heat-reclaiming device, the waste-heat boiler, which, in a measure, takes the character of an added engine auxiliary, although in some instances the boiler and muffler are built as a combined unit.

In order to work satisfactorily in conjunction with an internal-combustion engine, the waste-heat boiler, utilizing the exhaust gas as a source of heat, should have the following characteristics:

1. A good heat-transfer rate at the temperatures found in engine exhausts.
2. Resistance to corrosion from moisture and dilute acids.
3. Should be readily cleaned of carbon deposits and scale formations.
4. Good exhaust-muffling characteristics.
5. The boiler should be gastight.
6. Independent oil firing should be possible.
7. Minimum increase of exhaust back pressure.
8. Construction requiring a minimum amount of attention and maintenance.
9. Minimum of boiler accessories.
10. Low first cost.

While the foregoing requirements for waste-heat boilers specifically set forth resistance to corrosion, the D.E.M.A. in

their "Standard Practices" strongly recommend that exhaust-gas temperatures be kept above the dew point in order to minimize corrosion difficulties.

Waste-heat boilers on the market at the present time can be classified in the same general manner as conventional boilers, *i.e.*,

1. Fire-tube type.
2. Water-tube type.

In addition, the Clarkson[1] or "thimble-tube" boiler falls into neither of these classifications. Test data for a No. 8 Clarkson boiler with a 750-hp four-cycle mechanical-injection engine are contained in Table 51.

TABLE 51.—SUMMARY WASTE-HEAT BOILER TEST, CUSHING, OKLA.
Type engine: four cycle, 17½- by 25-in. cylinder, 225 rpm mechanical injection

Engine load			
Horsepower output	375	562.5	750
Pounds fuel per hp-hr, 19,270 Btu per lb	0.400	0.353	0.349
Btu input to engine per hp-hr	7,700	6,800	6,720
Exhaust temperature off engine	395	470	640
Pounds steam produced per hour	23	220	226
Pounds steam produced per hp-hr	0.061	0.392	0.302
Steam pressure (psi gauge)	2	4	7.8
Feed-water temperature at boiler inlet, deg F	98	93	92
Btu added per pound steam	1,089	1,096	1,099
Btu recovered per hp-hr	115	430	332

The test data in Table 51 are not complete in that temperatures of the exhaust gas entering and leaving the boiler are not given. Peculiarly these data show a greater heat recovery at three-fourths load than at full load, although the exhaust gas was 170 F higher at full load.

In order to approximate the quantity of heat recoverable from the exhaust of a diesel engine, Bradford and Clarkson[2] developed the formula

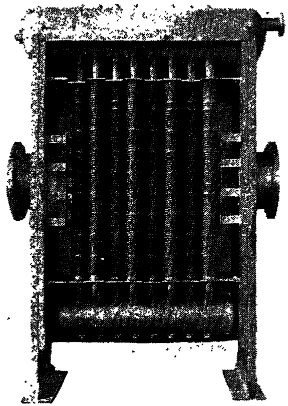
$$H = \text{bhp} \times C \tag{39}$$

[1] CLARKSON, THOMAS, and WILLIAM BRADFORD, Heat Recovery from Internal-combustion Engines, *Trans. A.S.M.E.* O.G.P. 53–54, p. 59, September–December, 1931.

[2] *Ibid.*

where H = Total heat recovered, Btu.
p = engine brake horsepower.
C = 12 for four-stroke-cycle engines.
= 20 for two-stroke-cycle engines.
D = temperature drop of gases.

When this formula is applied to the test data in Table 51, gas-temperature drops through the boiler appear to be somewhat erratic. This is probably due to the fact that the Btu recovery at three-fourths load was greater than at full load.



Foster Wheeler boiler. (See Table 52.)

The Foster-Wheeler Corporation has developed the data in Table 52 showing the steaming capacities of their waste-heat boilers for both four-stroke and two-stroke-cycle engines, and at different steam pressures. This table shows the economical range of steam production from a diesel-engine exhaust, and is self-explanatory.

Since waste-heat boilers normally operate at relatively low steam pressures, the only boiler auxiliaries necessary are

Sight gauge glass.
Safety valves.
Feed inlet valve, preferably automatic.
Boiler blowoff connection.
Indicating steam pressure gauge.
Gas pressure indicator to engine back pressure.

TABLE 52.—WASTE-HEAT BOILER-PERFORMANCE DATA

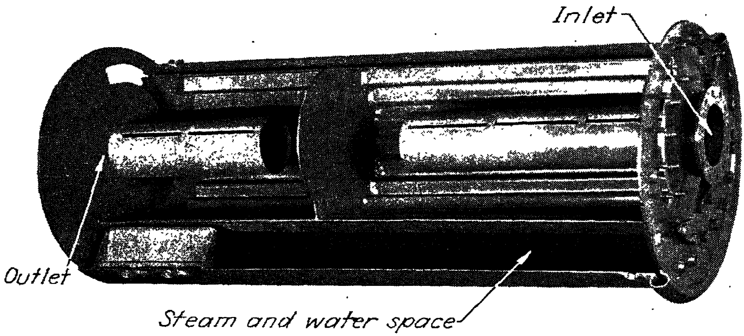
Hp	Lb steam per hr, full load			Lb steam per hr,			Lb steam per hr,			Sq ft heating surface
	5 psi	10 psi	15 psi	5 psi	10 psi	15 psi	5 psi	10 psi	15 psi	
Four-cycle diesel engines										
75	77	74	72	60	58	56	44	42	40	72
100	92	89	87	72.5	70	67	53	50.4	48.5	72
200	154	150	147	122	118	115	90	85	81.5	72
300	212	205	199	167	160	156	122	116	111	72
400	327	316	309	257	248	240	188	178	171	144
500	392	380	372	310	298	290	226	215	205	144
600	483	466	455	380	366	354	277	265	253	192
700	550	530	520	432	417	404	317	301	287	192
800	638	617	600	504	481	468	368	348	334	240
900	700	675	664	550	530	515	414	383	366	240
1,000	780	750	735	612	588	572	450	425	416	288
Two-cycle diesel engines										
75	63	60	57	40	37	34	17	14	12	72
100	78	74	71	49	45	42	21	17	14	72
200	165	157	150	102	94	88	44	36	30	144
300	229	217	208	144	133	124	60	50	42	144
400	300	285	273	191	176	165	80	66	55	192
500	370	351	336	234	216	202	98	80	68	240
600	450	427	410	282	260	244	118	98	82	288
700	525	498	477	330	305	285	139	115	96	336
800	600	570	545	378	358	320	158	130	109	384
900	675	640	615	425	393	368	178	147	123	432
1,000	750	710	682	475	440	410	198	163	136	480

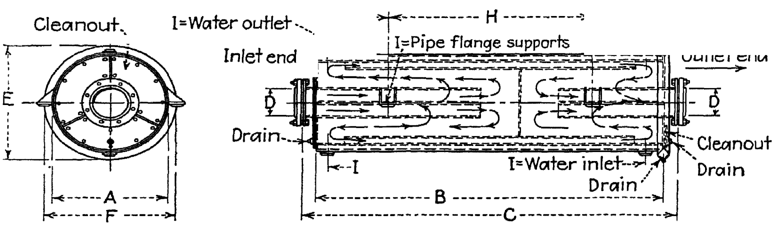
Basic data:
Gas flow
4 cycle = 12 lb per rated hp
2 cycle = 20 lb per rated hp

Feed water, 140 F

Load	Gas temperature, deg F	
	4 cycle	2 cycle
Full	700	500
Three-fourths	600	400
One-half	500	300

Courtesy of Foster Wheeler Corporation.





DIMENSIONS IN INCHES

Size	*A*	*B*	*C*	*D*	*E-F*	*G*	*H*	*I*	Estimated weight, lb	
									Dry	Wet
3-30	14	42	48	3	16	10	22	1	275	330
3¼-40	15	46	52	3½	17½	12	22	1¼	315	375
4-50	16	48	54	4	18½	12	24	1¼	410	475
5-70	20	58	64	5	23	14	30	1½	600	700
6-90	22	72	78	6	25½	18	36	1¾	800	940
8-145	26	87	93	8	30	22	43	2	1,200	1,400
10-210	34¼	97	104	10	38¾	24	49	2¼	2,000	2,300
12-300	40¼	117	124	12	45¼	30	57	2½	2,750	3,170
14-400	44¼	132	139	14	49¾	34	64	2¾	3,000	3,525
16-500	50¼	140	147	16	56¼	35	70	3	4,250	4,880

FIG. 148.—Standard sizes of combined heater and silencer, Maxim model WHS. (*Courtesy of The Maxim Silencer Company.*)

Where several engines exhaust to a common boiler, suitable gas dampers should be provided between the engine exhaust and the exhaust manifold to the boiler.

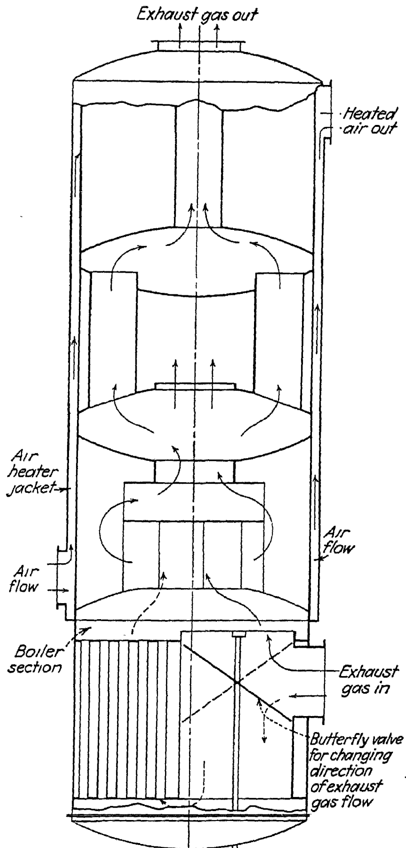


FIG. 149.—Heat-reclaiming, spark-arrester snubber. (*Courtesy of Burgess Battery Company.*)

In addition to waste-heat boilers, there are available today waste-heat water heaters where it is desired to obtain hot water from the waste heat in the exhaust rather than steam.

183. Combined Muffler and Boiler.—The two major manufacturers of mufflers have recently designed units that combine both the muffler and waste-heat boiler in a common unit. Maxim Silencer Company builds the unit shown in Fig. 148. The unusual feature of this unit lies in the fact that the steaming capacity is varied by changing the amount of water in the heat-recovery section. The greater the height the water is carried in the muffler-boiler, the greater the steaming rate.

An entirely different arrangement is that used by the Burgess Battery Company in their combined muffler boiler, Fig. 149. Here the boiler section may be either put into operation or by-passed through the operation of a butterfly valve within the unit.

184. Effect of Boiler on Exhaust Back Pressure.—Increasing the back pressure on a diesel-engine exhaust with a consequent reduction of the power output of the engine has long been a worry of the engine designer and operator. This effect was shown in Fig. 62. Some waste-heat boiler designers claim they can effect a reduction in exhaust back pressure by the use of a waste-heat boiler. On the other hand, others contend that a draft loss of about 3 in. of water is necessary through the boiler in order that the exhaust gas may have sufficient velocity to provide reasonable heat-transfer coefficients.

Installation records of four-stroke-cycle engines equipped with waste-heat boilers show that a back pressure of 3 to 4 in. through the boiler does not apparently affect engine economy or performance. While there is considerable difference of opinion as to the amount of back pressure a two-stroke-cycle engine will permit, a waste-heat boiler imposing as much as 3½ in. back pressure on this type of engine apparently does not adversely affect the engine operation.

185. Estimating Heat Recovery.—In order to approximate the probable heat recovery from an internal-combustion engine, Fig. 150 has been prepared. This chart, developed for mechanical-injection engines, permits a rapid check of the characteristics of exhaust gas for heating purposes. By starting with the rated engine horsepower the following items can be obtained in succession:

1. Quantity of exhaust gas per hour (which remains constant for all loads).

2. Total heat in the exhaust above 60 F, knowing the exhaust-gas temperature.

3. Total heat economically recoverable in Btu.

4. Total quantity of hot water or steam produced with inlet-water temperature of 60 F.

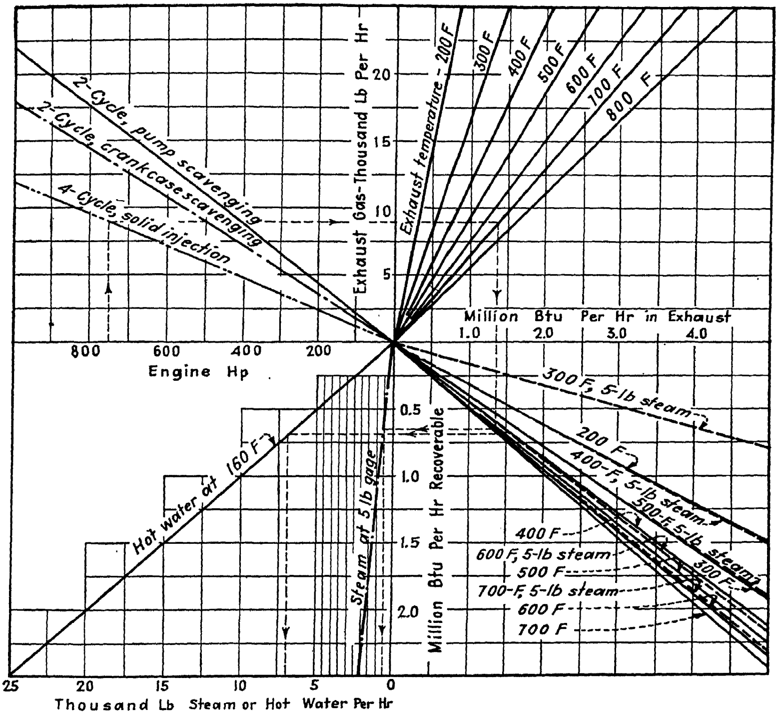


Fig. 150.—Chart for determining economical heat recovery from engine exhaust.

The example shown on the chart (Fig. 150) by means of the dotted lines and arrows is for a 750-hp four-stroke-cycle engine, considered to have an exhaust-gas temperature at full load of 700 F. The chart shows that approximately 8,000 lb of exhaust gas are available per hour carrying approximately 1,400,000 Btu above 60 F. The series of solid lines in the lower right-hand portion of this chart are to be used for determining the quantity

of heat economically recoverable in hot water, while the dashed lines are to be used for determining the quantity of heat economically recoverable in the form of steam at 5 lb gauge pressure. The temperature values carried by these lines correspond to the exhaust temperature under consideration; in this example 700 F. With a total of 1,400,000 Btu in the engine exhaust, 650,000 Btu can be recovered in steam at 5 lb gauge pressure or 700,000 Btu can be recovered in the form of hot water. Following the dotted lines through the lower left-hand chart, it is seen that approximately 600 lb of steam at 5 lb gauge pressure or 7,000 lb of hot water at 160 F can be produced.

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CHAPTER XVI

ACCESSORIES

In addition to the major equipment required for the functioning of any power plant, there are many auxiliaries that are required to ensure the proper operation of engines, generators, fuel handling, and other plant equipment. Many of these accessories have been discussed in other chapters. This chapter will be devoted to engine-starting facilities, meters, and instruments, and automatic-control equipment.

186. Engine-starting Systems.—Since manual starting can be used only for engines of small horsepower capacity, it is necessary to provide a power source for starting most engines. There are three methods used for engine starting as follows:

1. Electric storage battery.
2. Auxiliary gasoline engine.
3. Compressed air.

Some engines are equipped to start on low-compression pressure using gasoline for fuel, but these engines are largely confined to automotive service.

Storage batteries driving a starting motor in a manner similar to that used for automobile starting are employed on numerous high-speed diesel engines used in trucks, tractors, small power-generating units, and railway locomotives. For engines of 100 hp and smaller, a 12-volt storage battery is suitable, while larger capacity engines require a 24-volt battery. Batteries used for starting diesel engines must be specially designed to deliver large quantities of electrical energy during the starting period.

A small gasoline engine is used for starting the diesel engine on some makes of tractors, trucks, and railway locomotives, as well as on some engines driving electric generators. This auxiliary engine is usually cranked by hand. A clutch is provided which permits the gasoline engine to turn over the diesel engine until it begins operating, after which the clutch no longer functions.

All slow-speed engines used in central-station and industrial service are started on compressed air. Since most engines used in power-plant service require compressed air for starting, this starting system will be considered in detail.

187. Engine-starting Requirements.—Most engines employ 250 psi starting air pressure, although some require higher and some a lower pressure. The amount of air required to start an engine varies widely. It is influenced by the temperature of the cylinder walls and head, torque required to turn over the engine, oil drag, ignition characteristics of the fuel, skill of the operator in starting the engine, and many other factors. All other conditions being equal, a four-stroke-cycle engine generally requires more compressed air to start than does a two-stroke-cycle engine.

Since the amount of air required for starting is so indefinite, because of the many influencing factors, it is always advisable to provide ample capacity in air tanks to start an engine under all conditions.

188. Air-starting Equipment.—The equipment required for an adequate starting system using compressed air consists of one or two compressors, air-receiver tanks, and air piping, including the necessary safety valves, pressure gauges, and control valves.

If a single air compressor is used, it should be equipped with an electric motor and a gasoline engine to drive it. The gasoline engine is used for driving the compressor during the initial charging of the air receivers before the plant goes into operation and for emergency charging in the event the plant is shut down. After the plant starts operating, the electric motor drives the compressor for recharging the tanks, either through manual or automatic control. When two compressors are used, one is driven by a gasoline engine, and the other by an electric motor, Fig. 151. Both arrangements are used, although the arrangement using two compressors, while somewhat more expensive, is generally considered to be more reliable.

The number of air-receiver tanks installed is dependent largely upon the desires of the operator and the designing engineer. It is the usual policy when installing a single engine to provide two air receivers, while for installations having more than one engine, it is usual to provide one tank for each engine.

189. Air-starting Tanks.—Air-starting tanks may be constructed either by riveting or welding suitable steel plate. Owing to the high pressure involved, and the possibility of damage that might result from the failure of an air-receiver tank, all air tanks should be built in accordance with the A.S.M.E. Code for Unfired Pressure Vessels. It is a wise precaution to secure tanks designed for normal working pressures 30 to 50 per cent greater than the actual working pressure of the air-starting system.

FIG. 151.—Motor- and engine-driven air compressors for engine-starting system.

Openings should be provided in each tank for an air line, a drain connection, and a pressure-gauge connection. The main air connection may be either flanged or screwed, while drain and pressure-gauge openings are usually arranged for screwed connections.

Air tanks should be shut down and drained periodically to prevent corrosion on the inside tank surface.

There appears to be no standard range of sizes for air starting tanks. Diameters of tanks vary from 20 in. to over 4 ft, while tank lengths range from 5 to 10 ft and sometimes longer. The capacity of air receivers may be estimated by the use of Table 53 which gives the volume per foot of length of the receiver for

various diameters. No account is taken of the increase or decrease in tank volume for rounded or sunken heads.

TABLE 53.—AIR-TANK CAPACITIES

Diameter, Inches	Capacity in Cubic Feet per Foot of Length
18	76
20	2.18
22	2.64
24	3.14
26	3.68
28	
30	4.91
32	5.57
34	6.30
36	7.06
38	7.87
40	8.71
42	
44	10.58
46	11.52
48	12.58

NOTE.—To get the volume of a particular tank, multiply the length of the tank by the capacity per cubic foot for the particular diameter. For example a 30 in. diameter tank 8 ft long has a volume of 8 × 4.91, or 39.28 cu ft.

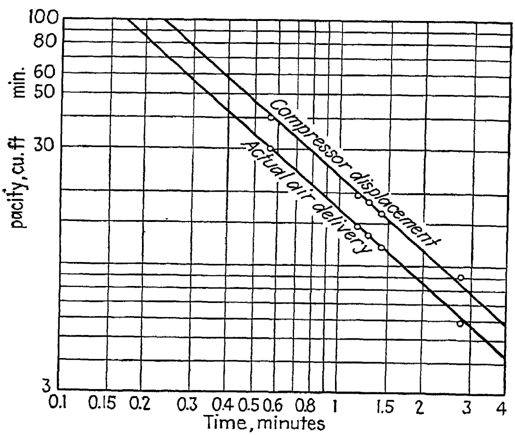


FIG. 152.—Chart for determining time required to compress 1 cu ft of air from atmospheric pressure to 250 psi gauge.

190. Air Compressors.—Compressors used for starting of internal-combustion engines are usually two-stage units equipped

for either motor or gasoline-engine drive or both. The size of an air compressor is determined primarily by the rate at which the air tanks are to be charged. The compressor used is usually small since the capacity of air tanks installed is limited. The amount of time the compressor operates is dependent upon the number of engine starts per day and the skill of the operator in starting the engine.

Figure 152 shows the time required to deliver a cubic foot of air at a pressure of 250 psi for various compressor capacities. The compressor displacement in cubic feet per minute and the actual air delivery are given. For example, a compressor having a displacement of 30 cfm will actually deliver 22 cfm of compressed air and requires 0.78 min to charge a volume of 1 cu ft from atmospheric pressure to 250 psi gauge.

Capacities of several sizes of compressors are given in Table 54. Different compressor manufacturers produce machines varying somewhat from these data, but the information is typical of the sizes and types of compressors usually found in internal-combustion-engine power plants.

TABLE 54.—AIR COMPRESSOR SIZES—TWO-STAGE UNITS DELIVERING AIR AT 250 PSI GAUGE

Size, in.	Displacement, cfm	Actual air delivery at 250 psi, cfm	Speed, rpm	Motor, hp
3½ × 2 × 2½	9.1	5.9	650	2
4 × 2½ × 3	16.5	12.0	760	5
4 × 2½ × 3	18.3	13.3	840	6
5¼ × 3 × 3½	19.3	14.5	440	7½
5¼ × 3 × 3½	40.0	30.0	900	15

191. Compressed-air Lines.—In determining the size of compressed-air lines required for connecting air tanks to the starting equipment on an internal-combustion engine, the nomographic chart, Fig. 153, will be found useful. With the quantity of air required by the engine known, the pressure drop for any size line can readily be determined.

192. Meters and Instruments.—The successful operation of any power-producing apparatus is dependent upon intelligent operation based upon knowledge of the condition of the equip-

ment at all times. Such knowledge is obtained only by using suitable meters and instruments. It is necessary, therefore, that the plant designer provide for the operating staff those meters

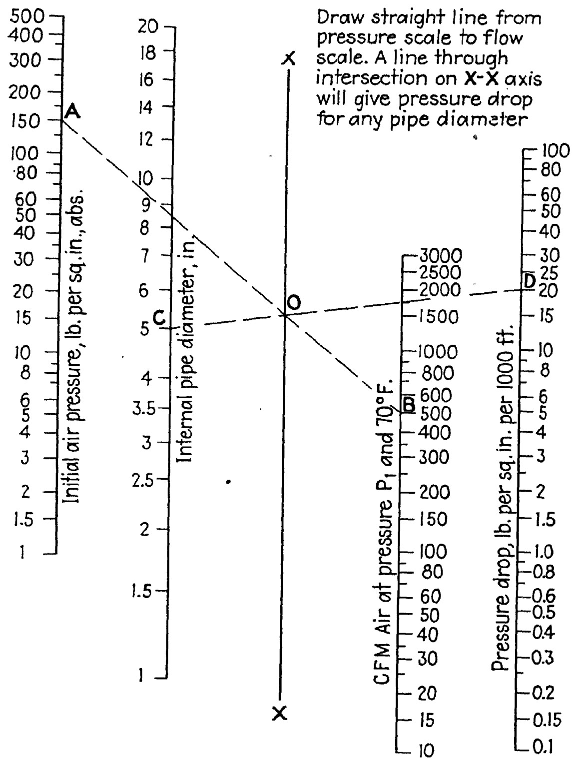


FIG. 153.—Nomographic chart for determining pressure drop in high-pressure air piping. (*Courtesy of Power.*)

and instruments which will aid in securing correct operating conditions.

193. Types of Meters.—Meters required for the operation of internal-combustion-engine plants can be grouped into three classes according to the functions of the meter. Practically all meters used will fall under one of the following three groups:

1. Meters indicating temperature.
2. Meters indicating pressure.
3. Meters indicating quantities.

Each of these groups can be further divided to include the particular service or condition requiring that type of meter. This division is as follows:

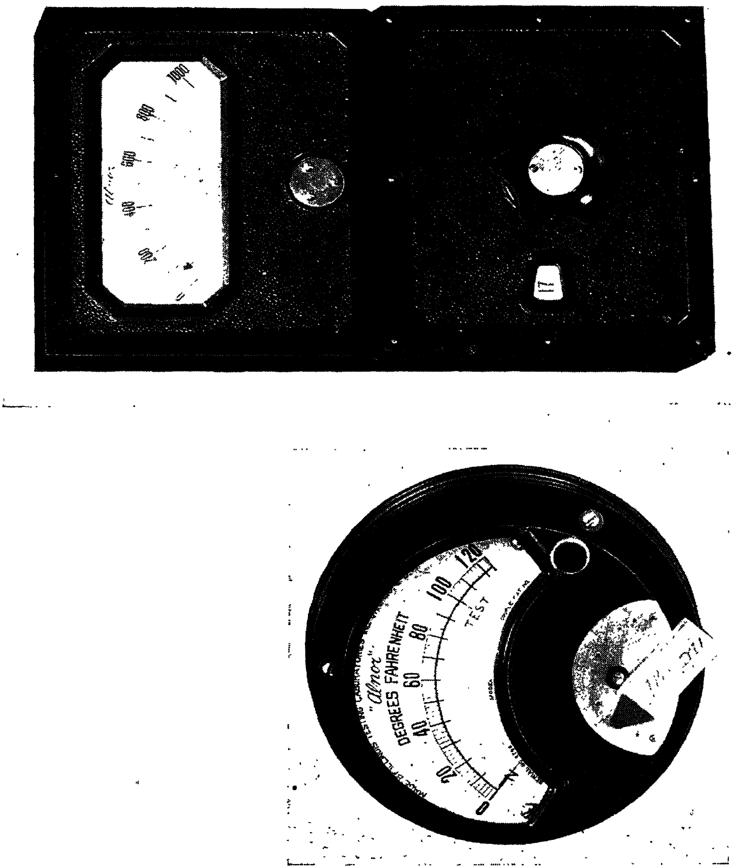
1. Meters indicating temperature.
   Cooling water in and out of engine.
   Cooling water in and out of lubricating-oil cooler.
   Lubricating oil in and out of engine.
   Exhaust of each engine cylinder.
   Fuel oil (particularly for heavy fuel oils).
2. Meters indicating pressures.
   Cooling water in and out of engine.
   Fuel oil to engine.
   Lubricating oil to engine.
   Starting air.
   Injection air.
   Scavenging air.
   Intake air.
   Suction and discharge of circulating water pumps.
3. Meters indicating quantities.
   Fuel oil delivered to day tanks.
   Water circulated through engines.
   Electrical energy produced.

All the meters set forth in the foregoing summary are generally of the indicating type. Where desirable, any or all of them can be made recording to provide a continuous record of the conditions at all metering points which may be used for further study or for checking conditions contributing to possible operating difficulties or failures.

194. Temperature Meters.—Temperature can be obtained by the use of thermometers, thermocouples, or optical pyrometers, although thermometers and thermocouples are the only ones employed in internal-combustion-engine plants. Mercury thermometers are used extensively for indicating temperatures of circulating water, lubricating oil, fuel oil, and for other temperature measurements usually below 300 F.

The thermocouple is universally employed for obtaining exhaust temperatures. The element is usually an iron-constantan combination, although nichrome-constantan, chromel-alumel, or nichrome-alumel may be used. Iron-constantan and

nichrome-constantan elements are good for temperatures up to 1600 F, while the chromel-alumel and nichrome-alumel elements can be used for temperature measurements as high as 2000 F.



B

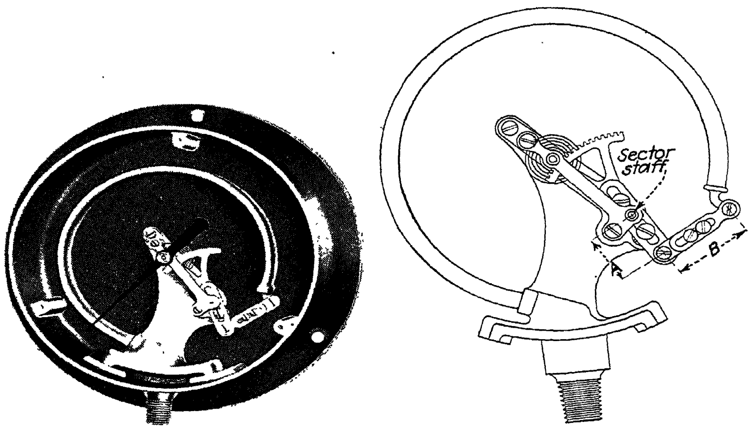
A

FIG. 154.—Pyrometer instruments used for measuring engine-exhaust temperatures. *A*. Round-type pyrometer. *B*. Rectangular-type pyrometer. (*Courtesy of Illinois Testing Laboratories.*)

The thermocouple is an electric battery, in which the voltage developed is proportional to the temperature difference between the cold end of the element and the point of application of heat

to the junction of the two dissimilar metals. A millivoltmeter, calibrated to read in terms of temperature, is used with the thermocouple. It is accurate to within ± 1 per cent of the full-scale reading of the millivoltmeter. Where a potentiometer is used instead of the millivoltmeter, the accuracy can be brought to within ± 0.05 per cent. For power-plant service, this extreme accuracy is not warranted and the millivoltmeter is universally used.

195. Pressure Meters.—Wherever pressure readings are required, it is customary to use bourdon tube gauges for this



A B

FIG. 155.—Bourdon-tube pressure gauge. (*Courtesy of Crosby Steam Gage & Valve Company.*)

purpose. They can be obtained for indicating a wide variety of pressures ranging from the extremely high to subatmospheric. The operating element in this type of gauge is a curved tube, one end of which is open to admit liquid, the other end closed. As the pressure on the liquid in the tube is increased, it tends to straighten out. This tube movement is transmitted through a link-and-ratchet assembly to the moving pointer which in turn indicates the pressure on a graduated scale. The internal workings of such a gauge are shown in Fig. 155.

For pressure measurements involving slight pressure drops, a simple U-tube arrangement, similar to that shown in Fig. 116,

may be used. For small pressure variations from atmospheric, water can be used in the tube, while mercury serves best for pressure variations of considerable magnitude.

196. Metering Quantities.—It is the general practice in internal-combustion-engine power plants to use displacement meters for measuring the quantity of oil received by the plant and delivered to the several engine day tanks. Where gas fuel is used, either an orifice meter or a suitable displacement meter is required. Water delivered to the cooling system on an engine can be measured by means of a suitable orifice or Venturi tube, or by means of a Rotameter developed by Schutte & Koerting Company for direct reading of the rate of flow.

Electrical meters and instruments are dealt with in considerable detail in Chap. XVIII.

197. Automatic Control.—Automatic control of internal-combustion-engine plants has received considerable attention, and some plants have been installed in the United States which are completely automatic. Applying automatic control to any installation is purely a matter of economics and reliability. Usually in a central station supplying electrical energy to a community, a staff of operators is required to handle certain functions that are not readily adaptable to automatic control. However, there are many operations in these stations which have a full operating staff which are entirely automatic in operation.

When the problem is studied in detail, it is seen that automatic control is used to some extent in many plants. As the number of automatic controls increases, the point is reached where the operation of the entire plant becomes automatic. There is a wide divergence between the plant in which only a few operations are controlled automatically and the installation where all functions are automatic in their operation. Essentially, automatic control can be divided into two groups of operations: those which are necessary to put the plant in operation and those required to shut it down.

198. Automatic Starting Devices.—Those devices necessary to start up an engine-driven generating unit and put it into operation must perform the following operations:

1. Start up the engine.
2. Adjust the engine speed.

3. Bring the voltage on the generator up to line voltage.
4. Bring the unit into synchronism with the line if the generator is an a-c machine.
5. Connect the generator to the line after the foregoing operations have been performed.

In starting up the engine, suitable interlocking electrical relays and electrically operated valves and controls must be provided to start up circulating-water pumps, open valves in the circulating-water system, start the engine on compressed air, and finally run the engine on its intended fuel. The other controls enumerated under items 2 to 5 have been used extensively in automatic electric substations for many years and are thoroughly reliable and dependable instruments.

199. Shutdown Devices.—In addition to starting up the plant, it is necessary that other instruments be available for shutting down a unit in the event that some mechanical or electrical difficulty develops. These shutdown devices are usually considered to cover the following:

1. Overspeed of the engine.
2. Failure of lubricating-oil supply.
3. Failure of cooling-water supply.
4. Electric generator failure.

It is customary in most of the plants now being built to provide some or all of these automatic shutdown devices even though a full operating staff is employed in the station at all times. This trend results from the desire on the part of the plant superintendent to protect his equipment even though it might result in a momentary interruption to service. Realizing the seriousness of engine overspeeding, or the failure of the lubricating-oil or cooling-water supply, he prefers to shut down the engine automatically rather than take a chance on seriously damaging the equipment. Wherever such shutdown devices are employed, they should be operated periodically to ensure that they will function in case of an emergency.

200. Alarms.—Most plant operators desire that certain of the operations in their plant be equipped with alarms to indicate to them when dangerous operating conditions may exist. For example, gongs are often wired into temperature-indicating instruments to warn the operator of high jacket-water or high lubricating-oil temperatures. Some operators also want to

know the condition of their air starting tanks, and oftentimes they have a gong or howler arranged to sound in the event their starting air pressure drops below a predetermined value. Such alarms are generally used to indicate that a condition exists which must be rectified immediately or dangerous operating conditions will result.

CHAPTER XVII

MAINTENANCE

The adequacy of the design of any internal-combustion-engine power plant is reflected in the ease with which it is operated; the continuity of operation is directly proportional to the effectiveness of the maintenance program instituted when the plant is put into operation. In reality, efficient operation of any mechanical equipment is as much a matter of proper care of the equipment as it is successful starting and stopping machines and correct manipulation of electrical auxiliaries.

A new diesel engine, properly operated, is expected to produce power with high economy and reliability. As the engine continues to operate, wear occurs in bearings, pistons, cylinder liners, piston rings, cams and cam rollers, and in other places where two surfaces move with respect to one another. The efficiency of the cooling system may fall off gradually, and other items of plant equipment may slowly lose their effectiveness. Such changes occur very gradually, and it is difficult in the day-to-day operation of the plant equipment to realize that they are taking place.

201. Adequate Daily Records.—Such gradual wear can be determined only through the compilation of adequate daily operating records and a systematic check and inspection of the equipment. The extent of the daily operating record maintained is influenced by the size and number of engines installed and the character of the service they supply. Regardless of the size or number of engines operating, a daily record should be kept as a guide for the proper maintenance and reconditioning of the units.

For diesel-electric generating plants, which supply an electric utility, the operating record maintained should be much more extensive than the record kept of a single engine supplying power for a small industry. The former would require complete records every hour or every half-hour of cylinder exhaust tem-

peratures; cooling-water temperatures to and from each cylinder jacket; lubricating-oil temperatures; fuel, lubricating oil, and air pressures; electrical-load readings for each generator; and weather conditions outside the plant. The latter might require

Hour	Bus Volts	Total K. W.	Unit No. 1															
			AMPS.			K. W.	Pressure				Temperatures						Fuel Oil	
											Water		Oil		Cylinder			
			1	2	3		Scav. Air	Water	Oil	Oil	In	Out	In	Out	High	Low	Filter Pressure	
M																		
1																		
2																		
3																		
4																		
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FIG. 156.—Portion of daily station log sheet.

only that a record be made once every 4 or 8 hr of temperatures, pressures, fuel consumption, and load. In either case, an adequate record should be kept of equipment operation.

202. Plant Cleanliness.—A clean power plant is usually well run and well maintained; a dirty plant may be, although it seldom is, well run and well maintained. Strange though it may seem

to some, this relationship between plant cleanliness and good operation is very marked. The operator who allows grease and oil to accumulate on the floor, lets wiping rags lie wherever they conveniently land, and permits trash to accumulate is very likely to be an individual who never pays much attention to the story the exhaust pyrometer tells, or attempts to keep any definite maintenance schedule for his equipment. On the other hand, the operator who keeps his plant cleaned up is usually a methodical individual with a definite maintenance routine that leaves nothing to chance.

203. Scope of Maintenance Program.—The maintenance program should be all inclusive, embracing equipment and auxiliaries in the plant as well as structures for housing equipment. In a broad sense the equipment and structures that must be kept in good repair if successful operation is to be assured can be classified as follows:

1. Engines.
2. Engine-cooling system.
3. Fuel-storage and handling equipment.
4. Lubricating-oil conditioning and handling equipment.
5. Air intake and exhaust equipment.
6. Starting system.
7. Meters and instruments.
8. Electrical equipment.
9. Plant structures.

Each classification will be considered in the order given.

204. Engine Maintenance.—To be successful, engine maintenance must be based upon adequate operating records and performed by skilled mechanical personnel. It must be done at the right time and in the proper manner.

Many maintenance operations are determined on the basis of hours of engine operation. Such a schedule must be developed for the engines being operated, since parts requiring attention may vary with the make and type of engine. When an engine is installed, the manufacturer furnishes with it a book of operating and maintenance instructions. Such instruction material should be studied carefully by those responsible for the maintenance of the equipment in order that the engines be kept in first-class operating condition at all times. From such instructions, and experience with the engine, a schedule of maintenance operations

can be developed. An example of such a maintenance schedule is given in Table 55.

TABLE 55.—ENGINE-MAINTENANCE SCHEDULE

Item to be inspected	Recommended maximum time between inspections Months
Remove and clean pistons	12
Clean and regrind exhaust valves	6
Clean and regrind air-starting valves	6
Clean and regrind air-starting check valves	
Fuel-pump valves cleaned and reground and plungers examined	
Check tightness of spray valves each time engine is shut down	
Check camshaft gears or chains and adjust if necessary	
Examine and adjust governor	
Check and adjust wrist-pin bearing each time cylinder is removed	
Check and adjust crankpin bearings	
Alignment of main bearings checked when crankpin bearings adjusted	
Main bearings checked and adjusted	
Crankshaft alignment checked and adjusted	
Lubricating-oil tanks and piping cleaned	12
Lubricating-oil piping examined for leaks and loose joints	2
Lubricating-oil pumps examined	
Cylinder-head jackets examined and cleaned	
Cylinder jackets examined and cleaned	
Flush out crankcase	

[a] This time interval is dependent upon the condition of the jacket water used for cooling the engine.

There are other adjustments that must be made from time to time to ensure the correct operation of the engine. Many of these adjustments result from information obtained by instruments installed to indicate operating conditions. As an example, the fuel pumps and injection nozzles of the several engine cylinders are adjusted at the time the engine is put in operation to develop substantially the same horsepower output in each

cylinder. This is checked by the exhaust temperature of each cylinder, and adjustments are made so that the maximum variation in exhaust temperatures from the several cylinders at one particular load does not exceed 20 or 25 F. If the maximum variation in exhaust temperatures for the several cylinders at the load where they were previously balanced exceeds approximately 40 F, it becomes necessary to make adjustments in the fuel-delivery system to equalize the load between cylinders. The cylinder with the highest exhaust temperature is doing the most work, and conversely, the cylinder with the lowest exhaust temperature is doing the least work, all other things being equal. Unless care is taken to see that the load is evenly divided among the several engine cylinders, some may be overloaded when the engine is operating at full-load rating. Pull cards taken with an engine indicator also aid in determining the relative performance of the several cylinders.

Engine compression should be checked periodically with an indicator. The diesel engine ignites the fuel charge by the heat of compression, and this compression pressure must be great enough at all times to raise the temperature of the air above the ignition point of the fuel. The probable causes for variations in compression pressure are leaky valves, loose piston rings, low scavenging-air pressure in two-stroke-cycle engines, obstructions in the air suction line, or increased piston end clearance.

Often the operator is confronted with the difficulty of trying to determine the condition of main and connecting-rod bearings, which, from inspection, appear to be badly corroded. As pointed out by Willi,[1] surface discoloration of a bearing does not necessarily mean that the bearing has been corroded or that it is unsafe for further use. Surface discoloration may be removed with an ordinary ink eraser, or by boiling the bearing in a 10 per cent solution (by volume) of Oakite No. 29 for 15 to 30 min followed by a thorough scrubbing of the surface with a suitable mechanic's soap. After such treatment, a bearing that has merely discolored will appear as clean and bright as a new bearing.

Hair-line cracks in the surface of the bearing metal often cause concern to the operator. Such bearings are usually satis-

[1] WILLI, ALBERT B., Bearings for Diesel Engines, *Mech. Eng.*, vol. 64, No. 6, p. 439, June, 1942.

factory for service until approximately 20 per cent of the bearing surface is covered with cracks. If, for example, after 1,000 hr of operation, cracks appear on approximately 2 per cent of the bearing surface, the bearing can be estimated to have a life of 10,000 operating hours, with 9,000 hr of useful life remaining. This comes about owing to the fact that the progress in the formation of such hair-line cracks is roughly proportional to the hours of operation of the bearing.

205. Cooling-system Maintenance.—Cooling-system maintenance should be conducted on a definite schedule to ensure proper operation of the cooling facilities. Water temperatures to and from the engine should be recorded periodically on the station log, and any variation in these temperatures from normal should lead to an immediate investigation as to the cause for the change. In this connection, all thermometers used for indicating water temperatures in the cooling system should be checked periodically to ensure their accuracy.

Gauges used to indicate pressure in the cooling-water system should be checked at least once a year, and preferably twice a year, by means of a dead-weight tester to maintain their accuracy.

If a cooling tower is used, the basin should be drained twice a year and all dirt, leaves, and other debris removed which might clog strainers, piping and valves, or pumps. The use of a divided cooling-tower basin, Fig. 99, aids in this cleaning operation, since one-half of the basin can be taken completely out of service. If spray coils are installed in the tower, they should be valved to permit taking a portion of the coils out of service for cleaning or tube replacement while the remainder continue in operation.

Shell-and-tube heat exchangers forming a part of a closed cooling system should be shut down periodically for cleaning. While out of service, all tubes should be checked for possible leaks. The frequency with which shell-and-tube exchangers are cleaned will be influenced by the rate of accumulation of dirt and scale in the exchanger. This rate of dirt and scale accumulation can be determined from the record of jacket-water temperatures to and from the engine, since the fouling of the heat exchanger will cause both the inlet and outlet water temperatures at the engine to rise above their normal values.

Radiators used for engine cooling should be drained and cleaned periodically. If make-up water added to the radiator is high in carbonate and noncarbonate hardness, it will be necessary to remove the scale from the radiator periodically by means of a suitable acid solution.

All piping and valves in the cooling system should be inspected periodically for leaks and for correct operation of valves. Many valves installed in cooling-water piping are used very infrequently in the normal course of plant operation. Such valves should be tested periodically to ensure that they will be in working condition when they are required.

Circulating-water pumps should be dismantled and thoroughly inspected at least once a year. In some cases, mineral deposits will accumulate upon pump impellers, thereby changing the head-capacity characteristics of the pumps. All such built-up materials should be removed from the impellers. Wearing rings should be inspected for clearance, bearings should be inspected for wear, and shaft packing should be tightened or replaced as the inspection may indicate.

206. Fuel-system Maintenance.—Maintenance of the fuel system is essentially a matter of keeping tanks, strainers, and piping systems clean and eliminating all leaks from fuel containers and piping system.

In plants employing oil for fuel, it is essential that all bulk storage tanks be taken out of service periodically and all water and sediment accumulated in the bottom of each tank cleaned out. While the cleaning-out work is in progress, the interior of the tank should be inspected for possible leaks. This applies to steel tanks located above or below ground as well as reinforced-concrete tanks installed underground. Steel tanks should also be checked during the cleaning period for possibility of internal or external corrosion, and if corrosion is found, immediate steps should be taken to eliminate the cause of such corrosion. Special care should be taken with vertical tanks set flush with the ground surface, to see that corrosion does not attack the bottom of the tank. Steel tanks in exposed locations should be painted periodically to minimize extended corrosion.

Where heating coils are installed in fuel-oil storage tanks, they should be checked periodically to ensure that leaks have not developed in the heating coils. Where such heating coils use

jacket water for fuel-oil heating, it is necessary to ensure that oil does not get into the jacket cooling water.

The efficiency of fuel-oil pumps should be checked regularly. Since most oil pumps are of the gear type and act as positive displacement pumps, the amount of wear in the pump can be roughly checked by noting the time required to pump a given volume of oil. Pumps used for unloading tank cars into station bulk storage tanks can be checked by determining the time required to unload a tank car of oil into storage. Transfer pumps for handling oil from bulk storage into engine day tanks can be checked by determining the time required to fill an engine day tank. As a pump wears, the slippage is increased, and consequently the time to pump a given volume of oil is increased.

One source of difficulty with the fuel-oil system is joint leakage in exposed and buried lines. Oil leaks in exposed joints can be checked by thoroughly cleaning the joint and painting it with whiting. Any trace of oil will immediately show stain on the white surface. Leaks in underground lines are more difficult to detect, although careful checks of the quantity of oil pumped through a line and the quantity delivered at the discharge end of the line will enable the operator to determine fairly accurately the leakage occurring in buried lines. This is a simple adaptation of the method used by gas and oil pipe-line companies to check leaks occurring in their cross-country transportation lines. Leaks in joints of a gas line can be detected by coating the joint with soapsuds.

Oil strainers should be cleaned periodically, care being taken to ensure that none of the foreign material caught by the strainer is permitted to get into the piping system beyond the strainer.

207. Maintenance of Lubricating-oil Equipment.—Maintaining lubricating-oil equipment consists essentially in keeping the reconditioning equipment in first-class operating condition and the tanks and piping used for handling lubricating oil clean and tight. The bulk of the lubricating oil used in the plant is reconditioned either with a centrifuge, filter, or chemical reclaimer. It is essential that this item of reclaiming equipment be kept in first-class working condition in order that purification of the lubricating oil will be satisfactory.

Centrifuges must be regularly cleaned of the carbon and other foreign matter removed from the lubricating oil. Heaters

employed with the centrifuge must be cleaned periodically or they will fill with sludge, and heating of the oil prior to centrifuging will not be satisfactory. Excessive wear of bowls and bowl disks which occurs over a long period of time may necessitate the rebalancing of the centrifuge-operating mechanism. This is usually a factory operation.

Filters should be cleaned periodically, the length of periods between cleaning depending upon the amount of foreign matter removed from the lubricating oil in relation to the capacity of the filter. Care must be taken in cleaning and recharging filters employing fuller's earth encased in cloth bags to see that no earth is permitted to get into the oil system and be carried over into an engine or damaged bearings may result.

Chemical reclaimers should be given a thorough overhaul at least once a year. Heating elements should be cleaned, thermostats checked to ensure correct operating temperatures, pumps checked for wear, and filter presses cleaned and inspected to ensure their correct operation. In operating a chemical reclaimer, care must be taken to see that fuller's earth does not get through the filter press and over into the oil being returned to the engine-lubricating system, or damaged bearings may result. In this connection, operators should be warned not to reuse filter paper in the filter press. Reuse of filter paper has resulted in fuller's earth being forced through the filter press, finding its way into the engine lubricating system, and damaging bearings.

Lubricating-oil storage tanks must be cleaned regularly. Those tanks holding only clean oil should be drained and cleaned out at least once every 6 months, and tanks used to hold dirty oil should be cleaned out at least every 60 days.

Lubricating-oil piping should be checked for oil leaks by the application of whiting to all joints as described in the discussion of the maintenance of fuel-oil piping.

Pumps used for transferring lubricating oil should be inspected regularly, and any wear in the pump compensated for in order that the pump function properly.

208. Air Intake and Exhaust Maintenance.—The maintenance of air intakes and exhaust mufflers is largely a matter of cleaning operations. Air filters must be cleaned whenever the pressure drop through the filter becomes so great that the engine cannot obtain sufficient air for proper combustion. As pointed out in

Chap. XIII, Art. 146, a suitable draft gauge should be installed on each engine air-intake line to provide the operator with a constant indication of the draft loss through the air filter and engine air-intake line. When the draft loss exceeds a value determined for the particular engine operating, it becomes necessary to clean the air filters to ensure an adequate air supply to the engine.

Air filters of the oil-bath type will not show so great a loss in air pressure through the filter due to dirt accumulation as will those of the viscous-impingement or dry type. It is necessary, therefore, with oil-bath filters to make a monthly inspection of the sludging of the oil in the filter to determine when it is necessary to clean out the filter and replace the oil in it.

Exhaust mufflers and exhaust piping should be kept free of carbon deposits. Excessive use of cylinder lubricating oil and poor combustion of the fuel in the engine will cause the formation of considerable carbon in the exhaust line and muffler of an engine. This carbon, together with such oil as might find its way into the exhaust system, is a potential fire hazard. Steps should be taken to ensure that an excessive amount of carbon does not build up in the exhaust line and muffler. In the event carbon does accumulate, it must be removed in order to eliminate the possible fire hazard and probable damage to the muffler and piping.

209. Starting-system Maintenance.—Maintenance procedure for the engine-starting system is influenced by the type of system employed. Where electric storage batteries are used, such batteries must be kept supplied with distilled water and charged in order to ensure proper operation of the starting facilities. Gasoline engines used for starting purposes must be operated regularly to ensure that they will function when required.

Air starting equipment and the connecting piping must be inspected regularly. Pipe joints should be checked for leaks by the use of soapsuds, valves should be operated to ensure that they are tight, and safety valves should be checked periodically to ensure that they will open within the pressure range for which they are set. Air-pressure gauges should be tested periodically with a dead-weight tester.

Air receivers should be drained and inspected for corrosion, both internally and externally, at regular intervals. Drains in

air piping lines should be opened periodically to remove all trapped moisture.

Air compressors should be given a regular overhaul in which bearings, rings, V-belt drives, motor and gasoline engine should be checked for wear and satisfactory operation.

210. Maintenance of Meters and Instruments.—All meters and instruments including thermometers, pressure gauges, and electrical meters should be periodically checked against known standards to ensure their accuracy. In a recent instance, a diesel engine driving an electric generator was severely overloaded because the wattmeter on the switchboard was not registering correctly. What the operator took for full load as measured by his indicating wattmeter on the switchboard was substantially 40 per cent overload, the error being due to a blown potential fuse causing the wattmeter to read incorrectly.

All meters and instruments are delicate mechanisms and can be damaged beyond repair through careless handling. Owing to the fact that these instruments so vital for the successful operation of the plant are delicate, it is advisable to have one man on the plant staff responsible for the maintenance and checking of all meters and instruments. In most electric utilities it is usual to have a man in charge of the testing and repair of watthour meters used for metering the electricity sold to consumers. This individual is usually the best qualified to test, inspect, and repair thermometers, pressure gauges, and electrical instruments, and he should be charged with the responsibility for their correct functioning.

Operators should be impressed with the fact that thermometers, pressure gauges, and electrical instruments are delicate, and should be cautioned against damaging them through carelessness or inexperience in making adjustments on them.

211. Maintenance of Electrical Equipment.—The maintenance of electrical equipment deserves as much care and attention as does the maintenance of the engine and its mechanical accessories. Cleanliness in electrical equipment will pay large dividends in trouble-free operation of generators, exciters, motors, and electrical-control equipment. Oil, dirt, and moisture are the three elements that cause havoc with electrical equipment, and the removal of these three by careful cleaning will eliminate a large part of the potential difficulties with electrical equipment.

Electric generators, exciters, and motors should be cleaned at least once a month with compressed air. Switchboards should also be cleaned monthly to ensure that dust and dirt will not interfere with the correct operation of relays, meters, and instruments.

Motor-operated devices used for emergency service should be operated periodically to ensure that no damage has been done to the motor and controls while the unit was idle. A motor of this type may deteriorate from corrosion or moisture if not properly protected and tested.

Watthour meters should be cleaned, adjusted, and calibrated every 3 months. Relays and indicating instruments should be cleaned and calibrated at least every 6 months.

Oil circuit breakers should be inspected and cleaned at least every 6 months, or following six operations of the breaker under load. This cleaning and inspection should include testing of the oil to ensure that its dielectric strength is satisfactory, truing up electrical contacts that may have been burned in operation, and the adjustment of the breaker closing and opening mechanism to ensure that it works under all operating conditions. Air circuit breakers should be inspected at least once every 6 months to ensure that contacts are in good working order and that the breaker will open and close properly.

Special care should be taken to ensure that brushes on generators, exciters, and d-c motors are fitted properly. The correct type of carbon brushes should be used at all times, and they should work freely in the brush holders. In many stations, each shift operator is charged with the responsibility of inspecting all brushes at least once a shift to ensure that sparking, improper brush fit, or other brush defects do not occur.

The commutator on each d-c exciter should receive careful inspection to ensure the proper operation of the commutator and brushes, care being taken to ensure that no uneven wear, excessive brush sparking, or other damage occurs. The commutator must be cleaned at least once a month, and special precaution should be taken to remove dust and dirt from the space between adjacent commutator bars. After the exciter has operated for some time, it may be necessary to true up the commutator, undercut the mica insulation between bars, or both. These operations must be done carefully to ensure that a

satisfactory commutator surface is obtained. Special machines are available for truing up commutators and undercutting mica. A careful workman can do these maintenance operations without the aid of the special tools mentioned, but care must be exercised in performing the operations.

The brushes used on collector rings of the alternator rotor are often very hard, and excessive grooving and wear of the rings results. When collector-ring grooving becomes excessive, it is almost impossible to fit new brushes until the rings have been machined smooth. Brushes should not be too hard, and care should be exercised to ensure that irregular wearing of the collector rings does not occur.

212. Maintenance of Plant Structures.—Plant structures must be maintained properly to ensure that the machinery contained in the plant will be housed satisfactorily. This type of maintenance involves painting of structural steel and all exposed steel work to eliminate corrosion; replacement of broken and cracked window glass; maintenance of roofs, roof drains, and flashing to prevent leaks through the building roof; and the cleaning and painting of other structural items where such painting will prevent deterioration.

This structural maintenance is often neglected until a major repair or reconstruction is necessary in the structure. Lack of continuous maintenance is costly, and the failure to make repairs as they become necessary often results in excessive repair costs when a major repair results from the failure to correct the minor difficulties as they occurred.

CHAPTER XVIII

ELECTRICAL EQUIPMENT

The electrical equipment is of prime importance in any electrical generating station since the generator produces and the switchgear controls the energy produced in the station. This chapter will, therefore, be devoted to consideration of generators, switchgear, and allied electrical equipment associated with the production and control of electrical energy.

213. Selecting an Electric Generator.—The selection of an electric generator requires that a thorough study be made of the service to be supplied as well as the characteristics of equipment available to meet the conditions surrounding the particular application. In selecting a generator for a particular duty,

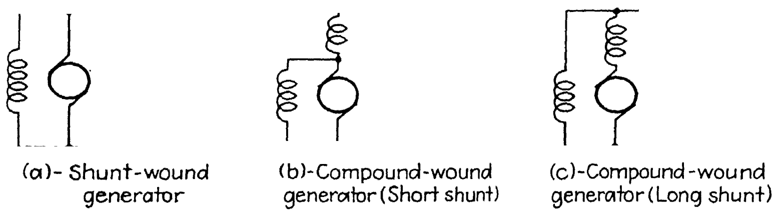

FIG. 157.—Direct-current generator connections.

the engineer should attempt, insofar as possible, to use standard equipment in order to keep the cost of the generator within reasonable limits. This requires that consideration be given to standards for generator rating, speed, voltage, temperature rise, frequency of alternating current to be supplied, and method of excitation.

Generators are available for the production of either direct or alternating current. In a majority of cases, a-c generators will be required since most of the electrical service throughout the United States is of this character. However, in some instances d-c generators will be needed, particularly where

electrical service is to be furnished in a hotel, office, apartment building, or industrial plant where d-c service is employed.

214. Direct-current Generators.—Two types of d-c generators, the shunt wound, and the compound wound, Fig. 157, are most widely used for power generation. Of the two, perhaps the compound wound finds the greatest application since by suitable proportioning of the shunt and series fields any voltage characteristic can be obtained. The voltage in a compound-wound generator can be made to fall, remain practically constant, or increase with an increase in load. On the other hand, a properly designed shunt generator operating at rated speed and within the range of load dictated by safe heating limits has an inherent tendency to regulate for nearly constant voltage.

Direct-current generators are usually built for full-load voltages of 125, 250, and 600 volts, although 275 volts is available for mining service. Standard sizes range from 1 to 2,500 kw

TABLE 56.—STANDARD SPEED RATINGS (RPM) OF ALTERNATING-CURRENT GENERATORS

Number of poles	Frequency, cycles per second		
	60	50	25
4	...	...	750
6	...	...	500
8	...	750	375
10	720	600	300
12	600	500	250
14	514	429	214
16	450	375	188
18	400	333	167
20	360	300	
22	327	273	
24	300	250	
26	277	231	
28	257	214	
30	240	200	
32	225	188	
36	200	167	
40	180		

with speeds from 80 to 1,750 rpm. Both two-wire and three-wire units may be obtained to meet conditions encountered on the system being served. Since voltages and speeds vary for the different generator ratings, it is advisable to consult the Standards for Motors and Generators of the National Electric Manufacturers' Association in selecting the d-c generator required for a particular installation.

215. Alternating-current Generators.—Alternating-current generators, usually referred to as alternators, for operating with diesel or gas engines, are built in standard sizes ranging from 1 to 8,000 kw, and for frequencies of 25, 50, and 60 cycles per second. Voltages range from 120 to 23,000 volts, although the more common are 240, 480, 600, and 2,400 volts. Rotating speeds of alternators most commonly used with internal-combustion engines for the three standard frequencies are given in Table 56.

While the range of speeds of engine-driven alternators most widely used in internal-combustion-engine plants is shown in Table 56, it is also advisable to know the standard sizes of generating units as well as the range of speeds and voltages available. This information, for units varying in size from 31 to 5,000 kva, is summarized in Table 57.

216. Generator Efficiency.—The efficiency of a generator is the ratio of the power produced by the machine to the power required to drive it. Stated in the form of an equation

$$\text{Generator efficiency} = \frac{\text{power output} \times 100}{\text{power input}}$$

where both input and output are measured in kilowatts. The difference between the input to the generator and its output constitutes the losses. Generator efficiency as determined by the American Institute of Electrical Engineers deals only with the generator proper and does not take into account the losses in the exciter and the field rheostat, neither of which is distinctly a part of the generator. Thus the only losses considered by the institute in its determination of generator efficiency are the following:

1. I^2R loss of armature and field coils at 75 C.
2. Core losses.
3. Friction and windage losses.
4. Stray load losses.

Table 57.—Summary of Alternating-current Generator Capacities, Speeds, and Voltages

Generator output		Speed range, rpm		Voltage			
Kva	Kw at 0.8 pf	Max.	Min.	240	480	600	2400
31	25	720	300	x	x	x	x
44	35	720	277	x	x	x	x
63	50	720	257	x	x	x	x
94	75	720	240	x	x	x	x
125	100	720	225	x	x	x	x
156	125	720	225	x	x	x	x
187	150	720	225	x	x	x	x
219	175	720	200	x	x	x	x
250	200	720	200	x	x	x	x
312	250	720	150	x	x	x	x
375	300	720	150	x	x	x	x
438	350	720	100	x	x	x	x
500	400	720	100	x	x	x	x
625	500	720	100	x	x	x	x
750	600	720	100	x	x	x	x
875	700	600	100	x	x	x	x
1,000	800	600	100	..	x	x	x
1,125	900	600	100	..	x	x	x
1,250	1,000	600	100	..	x	x	x
1,563	1,250	600	100	..	x	x	x
1,875	1,500	450	100	..	x	x	x
2,188	1,750	300	100	..	..	..	x
2,500	2,000	300	100	..	..	..	x
2,812	2,250	257	100	..	..	..	x
3,125	2,500	200	100	..	..	..	x
3,750	3,000	200	100	..	..	..	x
4,380	3,500	200	100	..	..	..	x
5,000	4,000	200	100	..	..	..	x

a.—Where lower voltages are not listed as standard, they may be obtained at an increase over standard price.

When efficiency information is secured for a generator to be driven by an internal-combustion engine, it is essential that the input-output efficiency of the machine including exciter and

TABLE 58.—PROBABLE INPUT-OUTPUT ALTERNATOR EFFICIENCIES

Generator output		Speed, rpm	Efficiency, per cent		
Kva	Kw at 0.8 pf		½ load	¾ load	4/4 load
125	100	225	82.6	85.2	86.7
		240	82.9	85.5	86.9
		257	83.5	85.9	87.3
		300	84.3	86.6	87.9
		327	84.8	87.0	88.3
		360	85.3	87.4	88.6
		400	85.9	87.9	89.1
250	200	225	86.4	88.3	89.5
		240	86.6	88.5	89.6
		257	86.9	88.7	89.9
		300	87.6	89.3	90.5
		327	88.0	89.6	90.7
		360	88.3	89.9	90.9
		400	88.9	90.3	91.3
625	500	225	89.9	91.1	92.0
		240	90.0	91.3	92.1
		257	90.2	91.4	92.3
		300	90.8	92.0	92.7
		327	91.0	92.2	92.9
		360	91.2	92.4	93.0
		400	91.5	92.6	93.2
1,250	1,000	225	91.7	92.8	93.4
		240	91.9	92.9	93.5
		257	92.0	93.0	93.6
		300	92.5	93.4	94.0
		327	92.6	93.5	94.1
		360	92.7	93.7	94.2
		400	92.9	93.9	94.4
1,875	1,500	225	92.6	93.6	94.2
		240	92.8	93.7	94.3
		257	92.9	93.8	94.4
		300	93.1	94.0	94.6
		327	93.3	94.2	94.7
		360	93.3	94.3	94.8
		400	93.6	94.5	95.0

rheostat losses is obtained and not the conventional A.I.E.E. efficiency. The difference between these two values at certain loads on the generator may exceed 2 per cent.

For the purposes of comparison, the input-output efficiencies for several sizes of engine-type alternators at various speeds are contained in Table 58. It is readily apparent that the speed of the generator as well as its size influences the efficiency. Alternators having higher efficiencies than those shown in Table 58 may be purchased at a somewhat greater cost than the machines conforming to these efficiency figures.

217. Capacity of Alternator Determined by Heating.—The capacity of an alternator is determined by heating of the stator and rotating field windings and is dependent upon the maximum temperature at which the insulation of the windings will continue to function satisfactorily. This maximum allowable or "hottest spot" temperature is in turn determined by the type of insulating material employed. Standards of the A.I.E.E. set forth the type of insulating materials as well as their hottest spot temperatures.

Class *O* insulating material consists of cotton, silk, paper, and similar organic materials when neither impregnated nor immersed in oil, and the maximum temperature at which they will function satisfactorily is 90 C. Class *A* insulating material consists of cotton, silk, paper, and similar organic materials impregnated or immersed in oil, and this class of insulating material will function satisfactorily with temperatures as high as 105 C. Class *B* insulating material consists of mica, asbestos, and similar inorganic materials in built-up form combined with a suitable binding cement and is satisfactory for hottest spot temperatures of 130 C. Only class *A* and class *B* materials are employed for the insulation of electric generators, although the class *A* material is generally employed for slow-speed machines driven by internal-combustion engines.

In actual practice, it is somewhat difficult to determine precisely this hottest spot temperature, and consequently it is necessary to make allowances in those temperature measurements which can be made in order to arrive at this value. When testing an alternator under load, two methods for determining temperatures of the windings are available. One is by the use of thermometers attached to the outside of the windings, and the second is by means of temperature detectors embedded in the

windings of the generator at the time of its construction and wired to suitable terminal boards on the generator frame. For most slow-speed generators of less than 1,500 kva capacity, temperature detectors are not used, and as a result thermometer readings only are available. Here again the A.I.E.E. has set up standard allowances for compensating thermometer readings in order to

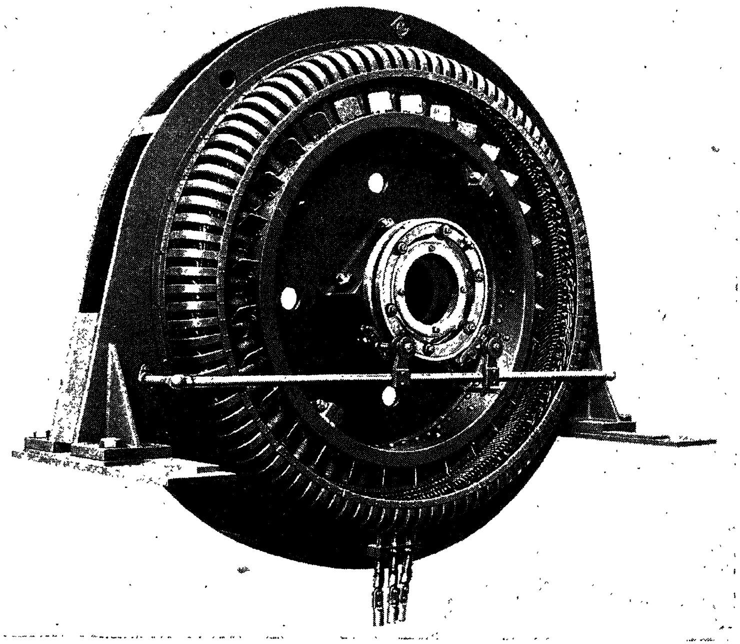

FIG. 158.—Engine-type alternator for operation with diesel or gas engine. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

obtain a close approximation of the hottest spot temperature of the generator windings. For class *O* and class *A* materials, 15 C is added to the thermometer readings to obtain the hottest spot temperature, while in the case of class *B* material, an allowance of 20 C is made.

The standard rating for slow-speed engine-driven alternators is based upon a 50 C temperature rise at full load in the stator

windings when using class *A* insulation and operating at 0.8 power factor. Any variation from these basic standards results in a special machine which usually carries an increase in the purchase price. This standard temperature rise of 50 C is based upon an ambient air temperature of 40 C (104 F) which was chosen after studying prevailing summer temperature conditions throughout the United States. Consider that the hottest spot temperature permissible for class *A* insulating material is 105 C. With an air temperature of 40 C, a correction of 15 C to be added to thermometer readings of winding temperatures, and a temperature rise of 50 C, we obtain a hottest spot temperature of $40 + 50 + 15 = 105$ C which is the allowable hottest spot temperature for this class of insulating material. In some sections of the United States, summer temperatures in generating plants exceed 40 C (104 F) for considerable periods of time. Where such conditions occur, it is not possible to employ class *A* insulating material and still have a hottest spot temperature of 105 C when a 50 C rise is experienced. Since class *A* insulating materials are generally used for engine-driven slow-speed alternators, it becomes necessary to operate at full load with a temperature rise less than 50 C under these conditions. In such an instance, a 40 C rise machine is used. With such a machine, it is possible to obtain full load from the generator with the ambient temperature in the engine room of 50 C (122 F).

A 40 C rise machine is more expensive than a 50 C rise unit, but it has the advantage of being capable of delivering 125 per cent of rated capacity for 2 hr with a temperature rise of 55 C when the ambient air temperature is 40 C.

218. Alternator Power Factor.—Occasionally the operator of an internal-combustion-engine power plant insists upon purchasing an alternator to operate at power factors of 0.6 to 0.7, hoping thereby to eliminate some of the operating difficulties experienced with derating alternators designed for 0.8 power factor in order to handle loads at the low power factors experienced. The only answer in most cases of this type is to improve the power factor of the load being supplied and not attempt to operate generators at power factors lower than 0.8. Methods for power-factor improvement are discussed elsewhere in this chapter.

219. Effect of Altitude on Capacity.—Altitude above sea level influences the capacity of an alternator. Basic designs for 50 C

rise machines cover units satisfactory for operation at elevations up to 3,300 ft. Elevations above 3,300 ft require modifications in generator design at an increased cost for the unit. This increase in cost as given by one manufacturer is as follows, based upon 50 C rise machines.

Elevation, Feet	Price Addition, Per Cent
Up to 3,300	0
3,301– 5,000	1
5,001– 9,900	5
9,901–15,000	10

Generators rated for 40 C rise are satisfactory for full-load operation up to and including 9,900 ft, although they carry no overload guarantees when operating at altitudes above 3,300 ft.

220. Parallel Operation of Alternators.[1]—When alternators driven by internal-combustion engines are to be operated in parallel, it is essential that the fluctuations in speed of the engine due to the combustion pressure in the cylinder be kept within very close limits in order that satisfactory parallel operation will result. This is accomplished by the use of a flywheel which is generally placed between the engine and the generator which it is driving. When an engine having six or more cylinders is used, and especially in the larger sizes of engines, it is found desirable in many instances to incorporate the necessary flywheel effect in the generator rotor. In effect, the flywheel becomes the rotating field of the generator, and the stator or armature of the generator is placed around the combined flywheel and field poles of the unit. Such an arrangement reduces the over-all length of the engine-generator combination somewhat.

Where generators are designed to incorporate the necessary flywheel in the rotor, two types of rotor construction have been used with success. One type employs an all-welded rotor in which the rim, center disk, ribs, and shaft hub are welded into a homogeneous assembly. The other design employs a heavy rim into which a spider is press fitted and each spoke is keyed to the rim. In selecting the type of construction of the rotor for a particular installation, it is advisable to consult with the engine manufacturer since it may have some bearing upon the critical

[1] WARMING, TROELS, Power Pulsation between Synchronous Generators, *Proc. O.G.P. Div., A.S.M.E.*, 1942.

speed calculations for the engine crankshaft to which it is attached.

221. Excitation.—Excitation of the alternator field is generally supplied by a small d-c generator driven from the shaft of the alternator either directly or by means of V belts or a silent chain. When a direct-connected exciter is employed, it revolves at alternator shaft speed, while V-belt- or silent-chain-driven units revolve at speeds greater than the alternator. The higher speed exciter together with the necessary pulleys, belts, and belt guard is usually a less expensive installation than the direct-connected exciter.

Some plant operators prefer the silent-chain drive, when an exciter is to be driven at a higher speed than the alternator. With either V-belt or silent-chain drive it is advisable that the drive have more capacity than theoretical calculations indicate is necessary to pull the exciter. These drives are subjected to shock loading, particularly during short circuits. In most instances this drive should be capable of handling two to three times the normal full-load torque requirements of the exciter.

Where internal-combustion engine-driven alternators supply systems subjected to widely fluctuating loads, it is desirable to provide means for bringing voltage back to normal as rapidly as possible following a sudden increase of load on the system. Since the rotating speed of the generator is slow, ranging in most instances from 180 to 400 rpm, the rate of voltage recovery is slow unless means are provided for speeding it up. This can be done if the speed of the exciter is sufficiently great or if a pilot exciter is employed to increase the excitation rate.

The high rotative speed of the exciter can be provided either by means of a belt- or silent-chain-driven unit or by the use of a separate motor-driven exciter operating at either 1,800 or 3,600 rpm. The use of a pilot exciter with the main exciter is applicable to installations where the size of the generating unit is 2,000 kw or larger. The function of this pilot exciter is to energize the field of the main exciter and thus assist in building up the exciter voltage at a more rapid rate than normal which in turn helps reduce voltage fluctuations resulting from sudden load demands on the alternator. When motor-generator sets are used for providing excitation, care must be taken to see that at least one generating unit in the station is provided with an

exciter in order that service will not be completely stopped in the event of failure of the motor-generator set.

The d-c excitation required by a generator at full load is dependent upon the capacity of the generator as well as the speed at which it operates. Thus, the slower the speed of a generator of a specific kilowatt capacity, the greater the excitation requirements. This is shown by Table 59.

TABLE 59.—EXCITATION REQUIRED BY ALTERNATORS

Generator output		Kilowatts excitation at full load 0.8 pf at the following alternator rpm						
Kva	Kw at 0.8 pf	225	240	257	300	327	360	400
125	100	6.2	5.8	5.5	5.0	4.4	3.9	3.6
156	125	6.7	6.4	6.0	5.5	5.1	4.4	4.1
187	150	7.4	7.1	6.8	6.0	5.5	5.1	4.5
219	175	7.5	7.5	7.2	6.4	5.8	5.3	5.1
250	200	8.3	8.2	7.8	6.9	6.3	5.7	5.4
312	250	9.3	9.0	8.6	7.8	7.6	6.4	6.0
375	300	10.0	9.9	9.5	8.6	7.9	7.2	6.7
438	350	10.8	10.0	10.0	9.3	8.5	7.8	7.4
500	400	11.6	11.4	10.9	9.9	9.2	8.4	8.0
625	500	12.8	12.7	12.2	11.2	10.4	10.1	9.1
750	600	14.1	13.8	13.4	12.3	11.6	10.7	10.2
875	700	15.2	15.1	14.6	13.4	12.6	11.8	11.2
1,000	800	16.3	16.1	15.6	14.4	13.7	12.9	12.2
1,125	900	17.4	17.1	16.6	15.0	14.7	13.9	13.2
1,250	1,000	18.6	18.2	17.7	16.4	15.7	14.8	14.2
1,563	1,250	20.0	20.0	20.0	18.8	18.1	17.3	15.0

NOTE.—For larger size units, it is advisable to apply to the manufacturer for specific data as to excitation requirements.

222. Individual Exciter vs. Exciter Bus.—There are two methods employed for furnishing excitation to a group of alternators. One method uses a separate exciter on each alternator with no electrical connection between the various exciters. The other method, while usually employing an exciter with each generator, also provides means for connecting all exciters to a

common bus with the generator fields receiving their excitation from this bus.

Of the two methods, that in which no connection is made between the several exciters is favored by most operators and designers, because it eliminates the possibility of plant shutdown through an exciter-bus fault, eliminates difficulties encountered in parallel operation of exciters, aids materially in adjusting voltage of the alternator for synchronizing, improves the efficiency of alternators, and permits the use of an individual voltage regulator on each generator for better voltage and wattless power control. The use of the exciter bus permits the use of a single-voltage regulator for controlling the voltage of several generators. While the exciter bus has been used extensively, it is rapidly being eliminated in the newer installations.

223. Voltage Regulators.—The voltage of a d-c generator or an alternator will vary with the load on the unit unless means are provided to maintain this voltage at a constant value. This regulation must be accomplished automatically, and several regulators have been developed to perform this function. All of them operate on the same basic principle, although each employs a different method for effecting voltage control.

When the voltage of the generator varies, the voltage regulator changes the resistance in the field of the exciter, and consequently the excitation, by such an amount that the generator voltage is restored to its desired value. If the generator voltage drops as the result of a load suddenly applied, the regulator cuts out resistance in the exciter field, increases the generator excitation, and reestablishes the desired voltage. On the other hand, an increase in generator voltage causes the regulator to add resistance in the exciter field, thereby reducing the amount of excitation and again restoring the generator voltage to normal.

Many different schemes are used to accomplish this regulation, and some of them will be discussed here.

224. Rocking-contact Voltage Regulator.—The use of a rocking contact motivated by a sensitive torque motor is the means for varying the resistance in the field of an exciter employed in the regulator shown in Fig. 159. The split-phase winding of the torque motor in this unit as built by Allis Chalmers is connected across one phase of the alternator through a potential trans-

former *PT*. The regulator resistor *G* is connected in series with the shunt field winding of the exciter.

Assuming the generator to be partially loaded, the contact segments will take up a position in which a part of the resistance is short-circuited. This position does not change so long as the voltage remains constant, because the electrical torque produced in the drum *C* is counterbalanced by the torque of the spring *F*. An increase in load tends to lower the alternator voltage, but the

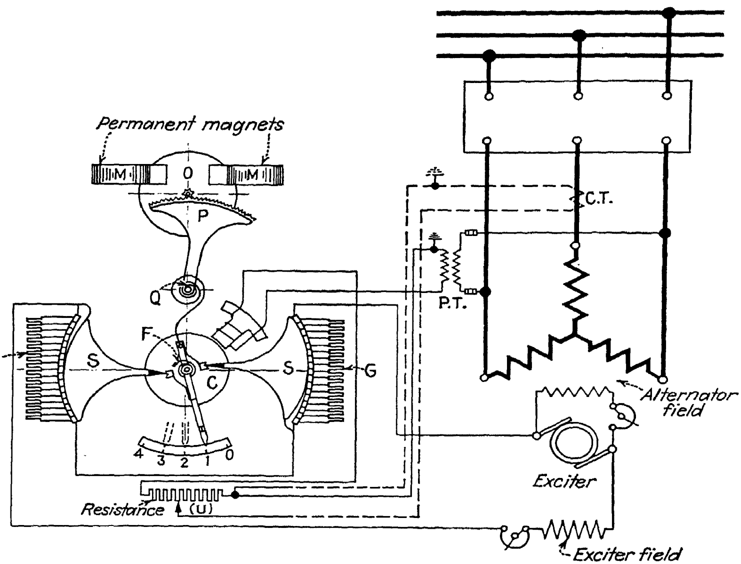

FIG. 159.—Schematic diagram of rocking-contact-type generator-voltage regulator. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

smallest decrease in voltage is immediately followed by a decrease of the electrical torque, so that the system becomes unbalanced. As a result, the sectors are displaced to the right till the mechanical torque, now increased by the torque of the recall spring *Q*, again equals the electrical torque. The resistance cut out of the field circuit by the displacement of the sectors is much in excess of what would be necessary to adjust the excitation to the new load condition. In this way the voltage of exciter and alternator

is forced to build up quickly, whereby the inherent inertia of the magnetic fields is partly overcome. In order to prevent overshooting of alternator voltage, this condition must change during the time taken by the alternator to respond to the new excitation value.

This is taken care of by the recall device and damping system in the following manner. The damping disk *O* is placed in the air gap of the two permanent magnets *M* mounted on a spindle, which is supported at both ends by jewel bearings. A pinion mounted on the damping disk spindle is geared to the rack of the damping sector *P* which is mounted so as to swing concentrically on the drum spindle and is connected to the drum by means of a flexible coupling (recall spring *Q*). Owing to the damping effect of the permanent magnets upon the rotating damping disk, the damping sector, although coupled to the drum, cannot follow as quickly as the sectors, and therefore they are displaced from their balanced position. The tension of the recall spring, however, is opposed to any displacement and will not only cause the damping sector to follow the movement of the sectors, but will tend to pull back these contact sectors, which have moved to a point equivalent to overexcitation of the alternator at the first instant of voltage drop. Thus pointer and damping sector will approach each other simultaneously from opposite directions and will come to rest when they meet, if at the same time the contact sectors are in a position equivalent to the correct excitation value for the new alternator load condition, and if the alternator voltage has returned simultaneously to normal.

225. General Electric Type GDA Regulator.—This regulator, Fig. 160, operates on the principle that any change in the a-c voltage causes the armature (8) of the voltage-sensitive torque element, which is normally balanced, to act against the tension of the spring (9) and assume a new position. This armature operates the rheostatic element (13) through the lever (10), altering the resistance the proper amount to correct excitation and restore the original voltage. The rheostat consists of two groups of resistance elements in series. Each element is provided with a silver contact button on the front end. As the lever (10) moves downward under the force of the spring (9), the front ends of these elements are successively brought into

contact, starting at the top, and the resistance of each element shorted out by the silver contact buttons.

226. Westinghouse Silverstat Regulator.—A third method for varying the resistance in the exciter field is employed in the Westinghouse Silverstat regulator, the operating element of which is shown in Fig. 161. The stationary coil is connected across one phase of the alternator through a potential transformer and a rectifier. A drop in alternator voltage reduces the

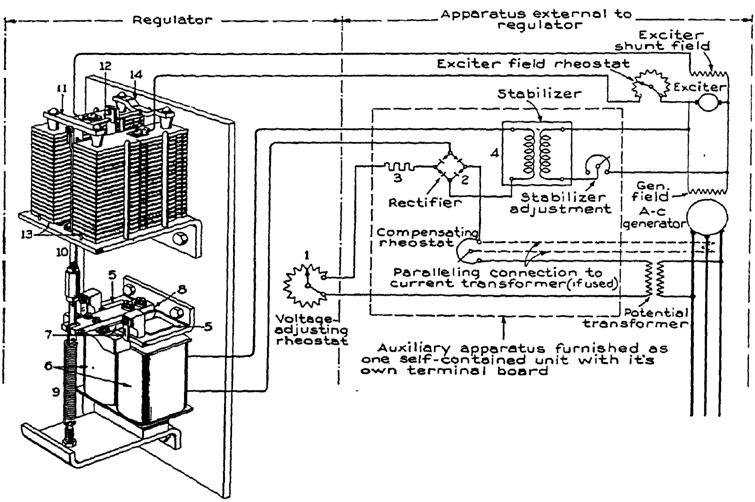

FIG. 160.—Perspective drawing and connection diagram of a two-stack, type GDA diactor generator-voltage regulator. (*Courtesy of General Electric Company.*)

current flow in the stationary coil, which decreases the magnetic field in the stationary magnetic circuit. This in turn causes the spring to pull the upper end of the moving arm to the right. As this occurs, the driving member at the top of the moving arm forces some of the individual leaf springs to make contact, shorting out a portion of the exciter field resistance. This causes an increase in the alternator field excitation, restoring the alternator voltage to the desired value. A rise in alternator voltage causes the top end of the moving arm to be displaced to the left, thereby introducing more resistance in the exciter field.

227. Telephone Interference.[1]—If a-c generators were to produce electrical energy having a pure sine current wave, any noise that might be created by induction in telephone circuits would not be objectionable. Unfortunately, currents in commercial generators do not follow pure sine waves. The actual current wave is that resulting from a combination of the fundamental frequency together with various harmonic fre-

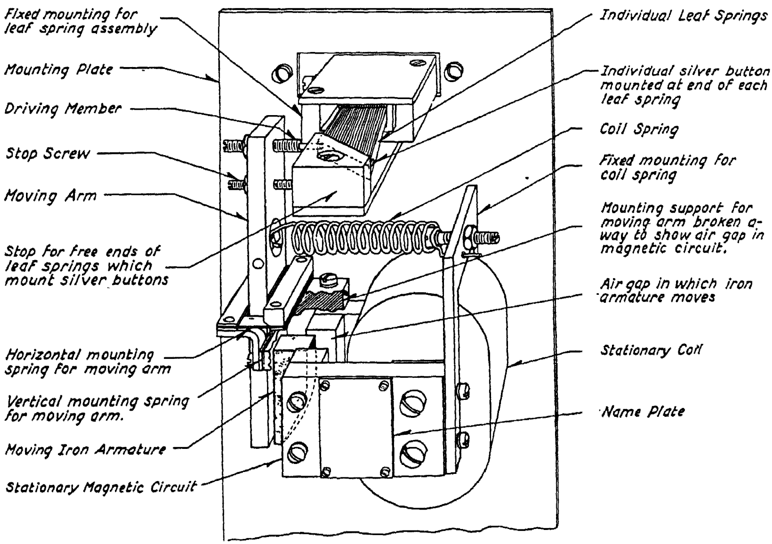

FIG. 161.—Diagram of construction of main control element of Silverstat generator-voltage regulator. (*Courtesy of Westinghouse Electric & Manufacturing Company.*)

quencies which are odd multiples of the fundamental. Those harmonic frequencies which occur within the range of voice and audible frequencies are the ones which cause objectionable noises in telephone circuits.

Engineers who have studied this problem have developed a term known as *telephone influence factor* (T.I.F.) which is a measure of the effect of either voltage or current in power

[1] CORBETT, LAWRENCE JAY, "Inductive Coordination," J. H. Neblett Pressroom, Ltd., San Francisco, Calif., 1936.

equipment and circuits on ordinary telephone equipment and telephone circuits due to wave shape only.

It has been recognized for many years that the design of generators had a great influence upon the harmonic frequencies that would be produced when the generator was in operation. In recent years the various manufacturers of electrical generating equipment have standardized upon maximum values of T.I.F. for various sizes of generating units. The present values are set forth in Table 60.

Table 60.—Maximum Values of T.I.F. for Engine-driven Alternators

Generator Capacity, Kva	N.E.M.A. Standard Max T.I.F.
62.5– 299	300
300 – 699	200
700 – 999	150
1,000 –2,499	125
2,500 –9,999	60

Generally speaking, it is desirable to secure as low a T.I.F. value as possible, particularly when a generator is supplying distribution and transmission facilities paralleling single-wire rural telephone or toll telephone lines. For that reason, telephone engineers in working with power-plant designers on interference problems have requested that generators be secured which have as low a T.I.F. value as is possible to obtain. In many instances systems have been provided with generating units having T.I.F. values below 75 for sizes smaller than 1,000 kva. While this is sometimes difficult to accomplish, and increases the cost of the generator, it has been found that the additional expenditure for generating equipment is warranted in many instances.

Each problem involving possible telephone interference must be considered in light of the particular conditions to be encountered and the difficulties to be eliminated. It is beyond the scope of this volume to go into detail regarding the many conditions that can cause telephone interference or the manner in which these interferences can be eliminated or reduced.

228. Circuit Breakers.—Circuit breakers of either the oil or oilless type are employed for the control of all circuits having a potential exceeding 600 volts. Generators operating at 2,400,

4,160, or higher voltages are connected to and disconnected from the system by means of breakers designed for the high-voltage high-capacity service to which they are subjected. All high-voltage circuits served from the system are likewise controlled by means of circuit breakers.

In the internal-combustion-engine power plant the breakers for connecting the generators to the bus act merely as a suitable means for connecting and disconnecting. Breakers for controlling feeder and station power circuits at the bus are equipped with automatic means for opening the breaker when a fault occurs on the circuit beyond the breaker. This breaker opening or "tripping" may be effected in several ways, although the most common method is to have the breaker open when an excessive flow of current occurs. Through the use of suitable control relays, breakers may be made to open through excessive current flow, a reduction in voltage, a reversal of current flow through the breaker, or by means of a variation in the current flow at two points in the circuit being controlled.

Until recently, practically all circuit breakers for a-c circuits in excess of 600 volts have been operated in an oil bath, the oil assisting in extinguishing the arc formed when the breaker contacts parted. In addition to oil circuit breakers, air circuit breakers have been developed for the higher a-c voltages and capacities.

For oilless circuit breakers having interrupting capacities from 50,000 to 150,000 kva, the arc extinction is effected by the use of magnetic or deion arc-quenching devices. For breakers having interrupting capacities exceeding 150,000 kva, compressed air is also used in some cases to assist in the arc-quenching operation.

In most internal-combustion-engine installations, breakers will seldom be needed for interrupting capacities exceeding 150,000 kva; therefore units of the oil or oilless type without the use of compressed air can be utilized successfully. There has been a desire on the part of some plant operators to get away from circuit breakers containing oil since there is a constant problem of maintaining sufficient oil in the breaker tanks, eliminating possible water contamination in the oil, danger from fire started in the breaker oil tanks, and excessive oil sludging and carbonization of the oil resulting from frequent operation.

Since both oil and oilless circuit breakers are available today for use in the power plant, and since both have proved themselves in actual field operation, the type of breaker selected will depend upon the preference of the man who must operate the equipment after it is installed.

229. Selection of Breaker Size.—Circuit breakers of either the oil or oilless type are used to protect electrical generating and other equipment from the serious effects of sustained short circuits. In order that the breaker perform its functions

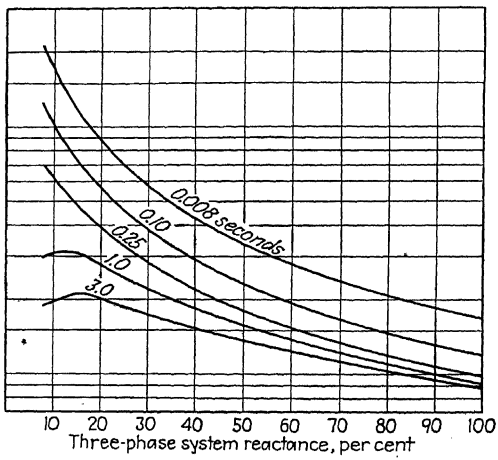

FIG. 162.—System short-circuit current (or kva) factors for three-phase short circuit on three-phase system.

satisfactorily, it is necessary that it possess suitable electrical and mechanical characteristics. The breaker must successfully interrupt the maximum kilovolt-amperes that it will be required to handle, and, in addition, it must possess sufficient mechanical strength to withstand the forces created by the initial current inrush during a short circuit.

At the present time, circuit breakers for 2,400- and higher voltage service are rated for kilovolt-ampere interrupting capacity and for 1- and 5-sec current-carrying capacities. These data can readily be obtained from any switchgear manufacturer. The standard ratings of oil circuit breakers for the ranges of

sizes usually employed in internal-combustion-engine power plants are given in Table 61.

TABLE 61.—INDOOR OIL-CIRCUIT-BREAKER CHARACTERISTICS

Rated			Short-time current rating rms total amp		Interrupting rating in 3-phase kva at rated voltage
Volts	Amp at				
	60 cycles	25 cycles	Momentary	5-sec	
2,500	60	60	3,000	3,000	10,000
	200	200	10,000	10,000	10,000
	300	300	15,000	15,000	10,000
	3,000	4,000	60,000	60,000	50,000
5,000	200	200	15,000	10,000	15,000
	400	400	15,000	10,000	15,000
	800	950	20,000	20,000	25,000
	2,000	2,250	40,000	40,000	50,000
7,500	400	400	20,000	20,000	25,000
	600	700	20,000	20,000	25,000
	1,200	1,400	35,000	35,000	50,000
15,000	600	700	25,000	25,000	50,000

To facilitate the determination of the proper breaker size, Fig. 162 has been developed by the manufacturers of circuit breakers. This is a series of decrement curves[1] which show the decay of the fault current produced by the alternator from the instant of application of the fault to the time the fault current reaches a sustained value. The application of these curves to actual cases follows.

Consider a three-phase 2,400-volt alternator of 500-kw capacity connected to a bus supplying two circuits, Fig. 163. Let us assume a short circuit adjacent to the terminals of one of the feeder breakers.

Known conditions:

Generator capacity 500 kw at 0.8 pf, or 625 kva.

Generator voltage = 2,400.

Reactance of generator = 22 per cent subtransient.

[1] HAHN, W. C., and C. F. WAGNER, Standard Decrement Curves, *Elec. Eng.*, vol. 51, No. 5, p. 324, May, 1932.

From Fig. 162, the initial inrush of current at 0.008 sec or the first one-half cycle for 22 per cent reactance is 8.0 times the full-load current. The initial current inrush can be calculated as follows:

$$\text{Current inrush} = \frac{625 \times 8.0 \times 1{,}000}{(2{,}400)\sqrt{3}} = 1{,}200 \text{ amp}$$

If we consider the breaker set to open in 15 cycles or 0.25 sec, and again referring to Fig. 162, it is seen that with 22 per cent reactance the current is now 3.55 times the full-load current. Therefore, the load that the breaker must interrupt is

$$625 \times 3.55 = 2{,}220 \text{ kva}$$

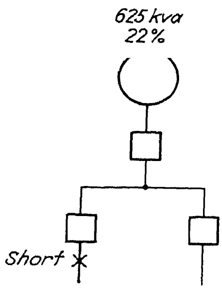

FIG. 163.—Short circuit on electric system with only one generator operating.

Referring to Table 61, it is seen that a 2,500-volt breaker having a 1-sec current-carrying capacity of 10,000 amp and an interrupting rating of 10,000 kva will be satisfactory under the conditions assumed. The smaller 2,500-volt breaker could not be used since its normal current-carrying capacity is only 60 amp., and the generator can deliver a total of $625 \times 1{,}000/2{,}400\sqrt{3}$ or 150 amp to either circuit.

In the foregoing example, reactance of all wiring from the generator to the fault has been neglected. This may be done unless it is desired to secure additional refinement of calculations. Usually such refinement is warranted only when large generating units are being considered and where the lengths of cable involved will affect the result appreciably. In general, circuit-breaker calculations for internal-combustion-engine power plants do not warrant such calculation refinements.

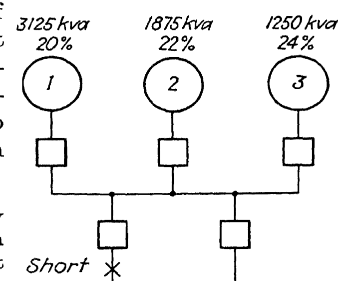

FIG. 164.—Short circuit on electric system with three generators operating.

The previous example involved only a single generator. Consider a station having three generating units of different sizes and subtransient reactances as shown in Fig. 164 operating at 2,400 volts, three phase with a short occurring at the terminals of a feeder breaker. We wish to know the size of the feeder breaker which will interrupt the power flow resulting from a short when all three generators are operating.

The reactance of each machine is given in per cent, and since the per cent reactance of each machine is different, it becomes necessary to determine an equivalent percentage value for the three machines operating in parallel. In making this conversion, the first step is to determine new values of per cent reactance for each machine on the basis of the total kilovolt-ampere capacity operating, and the second step is to calculate the per cent reactance for the units operating in parallel.

The new per cent reactance for each machine is determined by means of the equation

$$X_n = \frac{(\mathrm{kva}_t)(X_m)}{\mathrm{kva}_m} \tag{40}$$

where X_n = new per cent reactance of machine.
kva_t = total capacity of machine supplying the bus, kilovolt-amperes.
X_m = per cent reactance of machine.
kva_m = capacity of machine, kilovolt-amperes.

Returning to the example, Fig. 164, with a total of 6,250 kva connected to the bus, the new per cent reactance of each alternator is calculated as follows

$$\text{Per cent reactance No. 1} = \frac{6{,}250 \times 20}{3{,}125} = 39.1$$

$$\text{Per cent reactance No. 2} = \frac{6{,}250 \times 22}{1{,}875} = 73.4$$

$$\text{Per cent reactance No. 3} = \frac{6{,}250 \times 24}{1{,}000} = 150$$

Since the per cent reactance at the bus for the three units in parallel is the reciprocal of the sum of the reciprocals of the individual values calculated

$$\text{Per cent reactance} = \frac{1}{\frac{1}{39.1} + \frac{1}{73.4} + \frac{1}{150}} = 21.8$$

Again referring to Fig. 162 we find the inrush current for 21.8 per cent reactance to be 8.3 times full-load current, or

$$\text{Current inrush} = \frac{6{,}000 \times 8.1 \times 1{,}000}{2{,}400 \times 1.732} = 11{,}700 \text{ amp}$$

Likewise, the capacity that must be interrupted is

$$6{,}000 \times 3.6 = 21{,}600 \text{ kva}$$

when considering that the breaker will open in 0.25 sec (15 cycles).

Again referring to Table 61, it is seen that a 5,000-volt breaker having a 1-sec rating of 20,000 amp and an interrupting rating of 25,000 kva will be satisfactory. This breaker has a continuous current-carrying capacity of 800 amp, sufficient for the largest generating unit, which has full-load (full kilovolt-ampere) current rating of 750 amp. It is assumed that the current demands on any one circuit will not exceed the capacity of the largest

generating unit. If this is not the case, a breaker must be selected which will carry the current normally flowing through the circuit in addition to being able to handle inrush currents and interrupt the maximum short circuit possible.

In both examples given, the size of breaker required was determined for a particular condition. If the generating capacity in either case were to be increased materially, the breaker sizes selected would not be sufficient under the new conditions. It becomes necessary, therefore, in selecting breaker sizes to consider not only the immediate requirements for breaker capacity, but also the requirements for a sufficient period in the future to ensure that the breakers selected now will not have to be replaced prematurely. Additional interrupting capacity can be obtained very cheaply at the time the breaker is purchased, while the cost of substituting breakers possessing greater interrupting capacity within a period of 4 or 5 years may be excessive.

Where the capacity of existing breakers is being checked to determine whether or not they are adequate for additional generating capacity being installed, it is often advisable to employ more refined methods of breaker calculations than those set forth here. In such instances it is advisable to consult the manufacturer of the breaker.

230. Switchboards.—Switchboards for controlling the energy produced by electric generating plants are of two types, both being used extensively in internal-combustion-engine power plants. The conventional open switchboard employing panels of composition material or stretcher-leveled steel is still extensively used. In this type of construction, the panels are supported either by a pipe or welded angle-iron framework. Meters, instruments, and circuit-breaker control mechanism are located on the panel with the circuit breaker supported either on the back of the switchboard or on a separate framework apart from the board. In this latter arrangement, the breakers may be located at the rear of the switchboard or above or below the switchboard in a separate room or enclosure. A board of this type is shown in Fig. 165.

In order to increase the safety of high-voltage switching equipment as well as provide more compact and economical switchgear, the metal-clad type of construction has come into extensive use in recent years. The gear is essentially a series

of sheet-metal compartments for isolating items of operating equipment with the entire assembly forming one continuous housing when erected. Each generator is provided with a separate metal compartment comprising the panel on which are mounted the meters and instruments, a separate section for the high-voltage circuit breaker, another for current transformers, and still another in which the main bus is located. A similar arrangement is provided for each feeder circuit. Figure 166

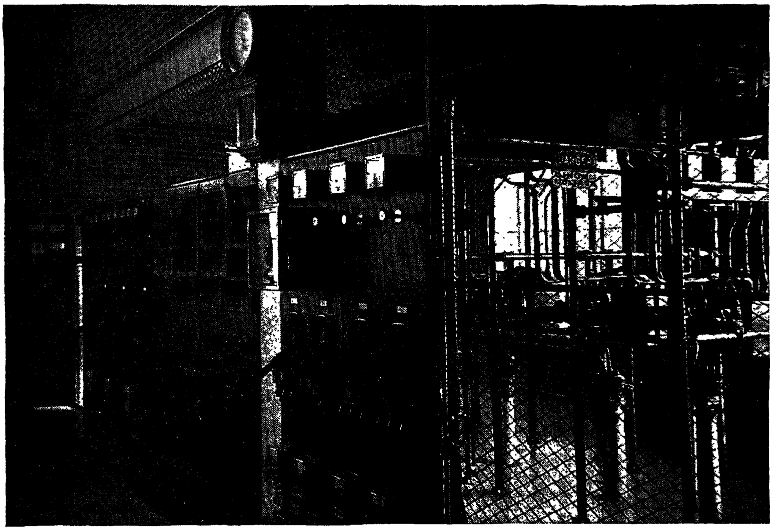

FIG. 165.—Conventional open-type switchboard. (*Courtesy of Burns & McDonnell Engineering Company.*)

is a photograph of an assembled metal-clad gear of this type, while Fig. 167 shows the method of installing or removing a breaker from a metal-clad switchgear section.

The use of metal-clad gear permits changes in and repairs to the various items of equipment without the attendant danger of contact with other high-voltage parts. Such repairs in the conventional switchboard construction, Fig. 165, cannot be made with such safety, since this barrier protection is entirely lacking.

FIG. 166.—Metal-clad switchgear installation for four generating units and seven feeder circuits. (*Courtesy of Burns & McDonnell Engineering Company.*)

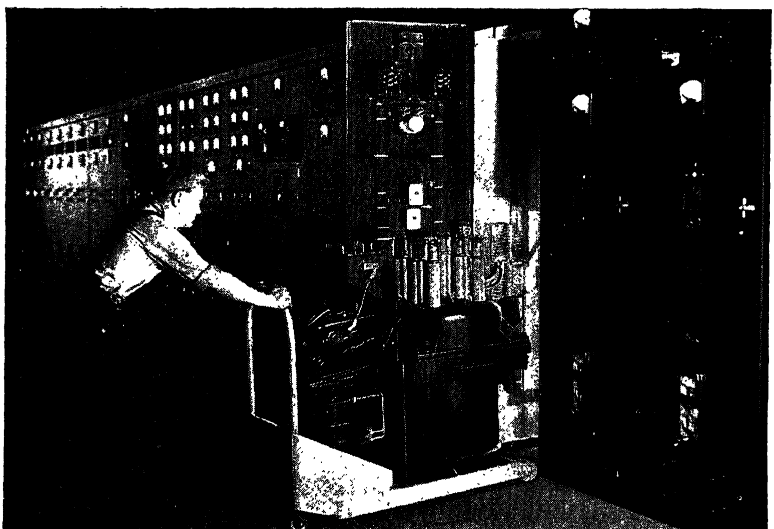

FIG. 167.—Placing breaker in metal-clad switchgear. (*Courtesy of Allis-Chalmers Manufacturing Company.*)

The use of metal-clad gear facilitates the removal of a breaker for inspection and repair and the substitution of a stand-by breaker in a matter of minutes. This eliminates the hazards common to so many conventional boards when the operator makes an inspection of the condition of the contacts and oil level in the circuit-breaker tanks. While this type of gear is more expensive than the conventional switchboard with separately mounted breakers, the installed cost of the two types is practically equal. This is due to the fact that the cost of the conventional type of gear does not include cost of installation, which is considerable since the board in many instances must be practically built in the field. On the other hand, the metal-clad gear need only be set in place on suitable supporting members and the wiring connections made to the gear, thus materially reducing the cost of field labor.

231. Switchboard Instruments.—In order to operate an electric generating station intelligently, it is necessary to know the conditions existing in the various electrical circuits. It is also necessary to operate a-c generating units in parallel. The procedure necessary for making these parallel connections requires the use of electrical metering equipment.

There are many combinations of electrical meters and instruments which can be utilized in the generating plant for checking the performance of equipment and electrical circuits. Each individual case may require different combinations of meters and instruments. It is possible, however, to enumerate those electrical control instruments normally used in an internal-combustion-engine power plant supplying a small community as a guide for those who are planning a new installation and for the operating man who wants to know the function of each instrument in the operating scheme. The following instrument and meter list is taken from an installation provided with a separate voltage regulator for each generator. If a single voltage regulator is used for the control of several generators it will be necessary to make some changes in the assembly of meters set forth in Table 62.

In the discussion of instruments just presented, it will be noted that polyphase watthour meters were provided only on the generator panels. The use of polyphase watthour meters on all the feeder circuits is a matter that must be decided for each

TABLE 62.—PLANT SWITCHBOARD INSTRUMENTS WITH INDIVIDUALLY REGULATED GENERATORS

Item	*Use*
	Synchronizing panel
1 Synchroscope	This instrument indicates whether the generator to be connected to the bus is slow or fast in relation to the machines supplying the bus and also the point at which the breaker of the incoming machine is to be closed. When a machine is connected to the bus it should be at the same frequency as the bus
2 A-c voltmeters	One voltmeter gives bus potential. The other gives the potential of the incoming generator. They must be equal
2 Synchronizing lights	These lights should be provided as a means of checking the synchroscope. The speed with which the lights become dim and bright indicates the difference in speed between the incoming machine and the bus
	Generator panel
1 Single-phase a-c ammeter and switch	This instrument together with a three-position switch permits reading the current in each phase of the generator. From these three readings the relative unbalance between the loads on each phase can be obtained
1 Field ammeter and shunt	This instrument shows the amount of current required for magnetizing the alternator field and indicates when an excessive magnetizing current and possible damage from coil heating may result
1 Polyphase indicating wattmeter	This instrument shows the instantaneous load on the generator in kilowatts
1 Polyphase watthour meter	This is an integrating-type meter and shows the total power generated by the unit in kilowatt-hours
1 Total hour meter	This instrument shows the total hours the unit has operated and is of value in assisting the operator in planning his engine-maintenance schedule
1 Governor motor control switch	A switch for varying the speed of the engine by regulating the engine governor necessary when synchronizing the unit with the bus
1 Power factor meter-transfer switch	If a power-factor meter is provided, this switch permits making a rapid check of power factor of each machine
1 A-c voltmeter switch	This switch is used in conjunction with one of the a-c voltmeters on the synchronizing panel to give the voltage of the generator for comparing with the bus voltage

Item	*Use*
1 Synchronizing switch	This switch is used for connecting the synchroscope to the machine while being brought into synchronism with the bus
1 Circuit-breaker control switch	This is either a manual breaker-closing mechanism or an electrical control switch depending upon whether the breaker is to be operated manually or electrically
	Totalizing section
1 Recording wattmeter	This instrument provides a graphic record of the instantaneous kilowatt loads on the station
1 Recording voltmeter	This instrument gives a continuous record of the bus potential and is a valuable aid in checking the efficiency of voltage regulators as well as the alertness of operators
1 Indicating frequency meter	This instrument gives a continuous visual indication of the frequency of the power being produced.
1 Indicating power-factor meter	This instrument is used in conjunction with generators for showing power factor and assisting in balancing circulating current. May also be wired to feeder circuits if desired
	Feeder section
1 Single-phase a-c ammeter and switch	This instrument is used to check current on each phase of feeder both for feeder load and unbalance between phases
1 Power factor meter-transfer switch	This is desirable where the plant serves a community, in order to determine those circuits needing power factor correction
1 Circuit-breaker control	This is either a manual or electric breaker-operating mechanism the same as for generator breakers
2 Control relays	Generally induction overcurrent relays are used to open the breaker in case of fault on the circuit. The type of relay used will usually be determined by the operating conditions involved

individual case, although it is highly desirable to have a polyphase watthour meter on the station power circuit. It is sometimes questionable whether a plant supplying a community should have a polyphase watthour meter installed on each outgoing feeder, although in an industrial installation, the metering of each circuit fed from the main switchboard might be of advantage.

Whenever an engine is equipped with shutdown devices that automatically stop the engine through any of the causes listed in

Art. 199, it is necessary to equip the generator driven by the engine with reverse power relays to open the generator breaker when the unit stops.

232. Station Bus Arrangements.—Many plans for power busses to facilitate the delivery of electrical energy have been employed. These schemes vary from the simple single-bus single-breaker arrangement, as shown in Fig. 168, to the most elaborate arrangement in the large power stations where all known methods of protection, control, and arrangement have been utilized to ensure continuity of power supply.

Because of its simplicity and low cost, the single-bus single-breaker arrangement utilizing only one breaker per circuit

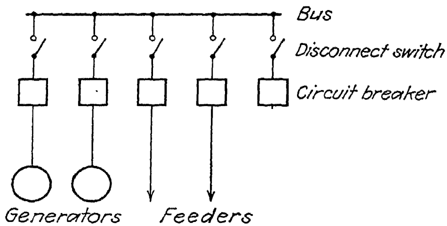

FIG. 168.—Single-bus, single-breaker layout.

and only a single bus is in common use in small power plants employing the conventional open-type switchboard and breaker arrangement. For the open type of switchboard construction, its main objection lies in the fact that no ready means is available for inspecting feeder breakers while deenergized, if the feeder is to be kept in service. Furthermore, the failure of one or more bus insulators would shut down the entire plant. This single-bus arrangement is not objectionable when metal-clad switchgear is used since special insulation is provided around the bus while any breaker can be removed and a substitute breaker installed in a matter of minutes.

In order to overcome some of the objections of this arrangement when employing the conventional open-type switchboard, the single-breaker double-bus arrangement shown in Fig. 169 has been developed. This scheme has the advantage over the single-bus arrangement in that a failure on one bus will not result in the station being shut down for the entire time necessary

to make repairs to the damaged bus. This system is more expensive than the single-bus arrangement and is open to the objection that feeder breakers cannot be taken out of service for maintenance in a deenergized condition. By the use of a transfer breaker in connection with the double-bus arrangement

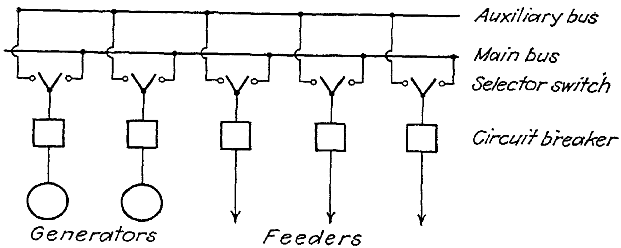

FIG. 169.—Double-bus, single-breaker layout.

in conventional open-type switchboards, as shown in Fig. 170, it is possible to overcome the undesirable maintenance features of the two previous arrangements.

When metal-clad gear is employed, the use of an auxiliary bus, as shown in Fig. 169, permits the repair of either bus in the event

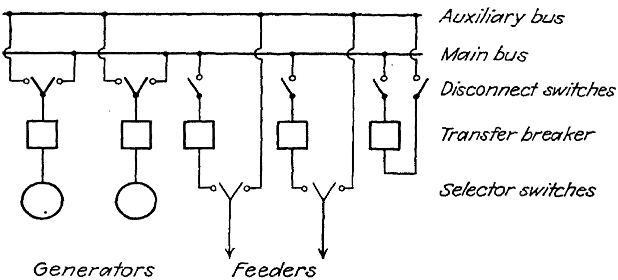

FIG. 170.—Double-bus, transfer-breaker layout.

that it should become damaged through electrical or mechanical failure. Maintenance can be accomplished on any breaker in the metal-clad gear while out of service, and any feeder breaker can be removed with only a very short interruption to service. Any generator breaker can be removed during such time as the generator is not in service.

A fourth and more elaborate scheme, which can be used either with the conventional open type of switchboard or with metal-clad gear, requires two circuit breakers for each generator and each feeder circuit connected to separate busses as shown in Fig. 171. This scheme is seldom used for internal-combustion-engine power plants since its high cost is generally not warranted.

233. Power-factor Correction.—In any a-c circuit the voltage and current are constantly varying in magnitude. In a 60-cycle system they are reversing their direction 120 times a second. The current and voltage may reach their maximum value in a given direction at the same instant, or they may reach their

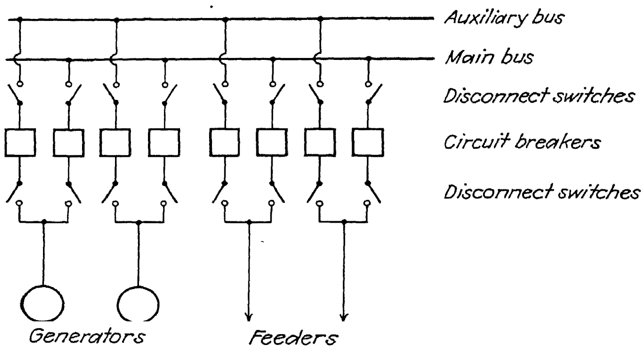

FIG. 171.—Double-bus, double-breaker layout.

respective maximum values in a given direction at different times. Whether the current and voltage reach their maximum value at the same instant or at different instants depends entirely upon the character of the circuit through which the electric current is flowing. When only resistance such as an incandescent light is present in the circuit, the voltage and current will reach their respective maximum values simultaneously. If the circuit consists of inductance and some resistance such as the iron core and coils of a transformer, the current lags behind the voltage and the amount of lag is dependent upon the relationship between the inductance and resistance present in the circuit. On the other hand, if only a condenser is present in the circuit, the current will lead the voltage.

In circuits where only a resistance, an inductance, or a condenser is present, the effect of the relationship between the current and voltage is as follows:

Resistance only..............	Current and voltage in phase
Inductance only..............	Current lags voltage 90 deg
Condenser only..............	Current leads voltage 90 deg

From the foregoing, it is seen that the effect of a condenser is just the opposite of an inductance and they tend to neutralize each other. Unfortunately, in actual practice, all the parts of an electric system tend either to possess resistance or inductive reactance. Induction motors, transformers, and even the spacing of the wires in an overhead distribution system contribute inductive reactance to the system. The neutralizing effect of a condenser can be secured only through the use of synchronous motors or condensers, called *capacitors*. In most small systems there are few synchronous motors used, and as a result it is necessary to provide the counteracting effect to inductive reactance through the installation of condensers where conditions require.

Induction motors operating at a fraction of their full horsepower rating contribute greatly to the inductive reactance of the system supplied. It is a common failing, too, in installing motors to pick them oversize in order to ensure that sufficient power will be available. By the proper selecting of motor sizes, power factor can be improved substantially in many instances.

When an internal-combustion-engine power plant is serving an electric distribution system, however, it is not possible to have complete control over motors installed on the system. The effect of badly lagging power factor due to inductive reactance loads of motors is very noticeable throughout the southwestern part of the United States where a large fan load for residence air conditioning is supplied during the night hours in summer. Since there is little or no load of other types to offset this inductive motor load in many communities, the power factor at the plant during these summer nights may be as low as 0.6 to 0.7. When such conditions occur, it is necessary that some remedial measures be employed to improve this power factor.

An illustration of what happens in the relationship between the voltage and current in a circuit due to the change in angularity

between the current and voltage is shown in Fig. 172. When the voltage and current are in phase, the power produced is a maximum. When the current lags behind the voltage, the power

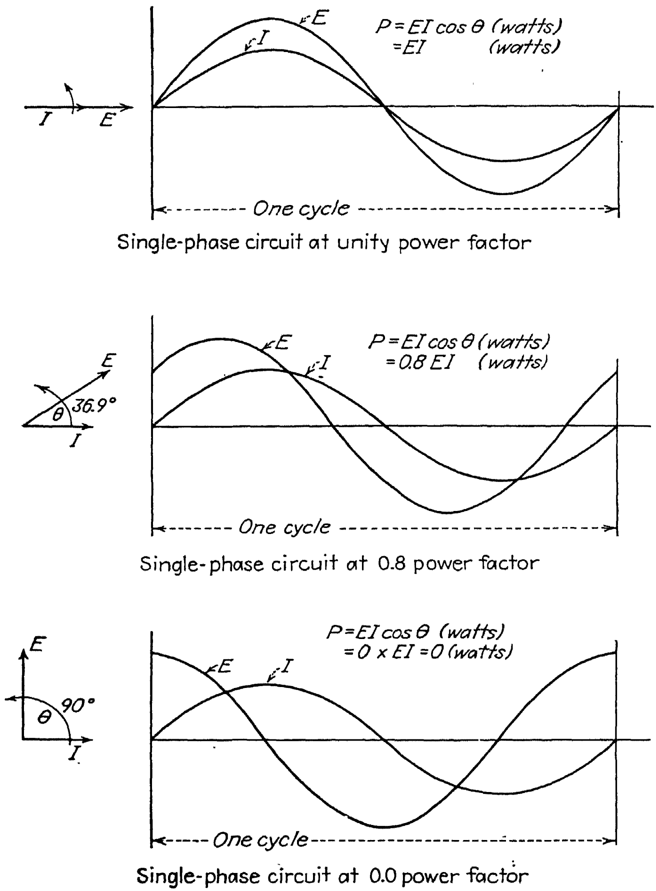

FIG. 172.—Effect of power factor on power delivered.

produced is less owing to the fact that the current and voltage do not reach their maximum values at the same instant. This variation is directly proportional to the cosine of the angle

between the voltage and current vectors. The power of a single-phase circuit, used for purposes of illustration, is given by the equation

$$P = EI \cos \theta \tag{41}$$

where P = power, watts.
E = voltage of circuit.
I = amperes flowing in circuit.
$\cos \theta$ = power factor of circuit.
θ = angular displacement between current and voltage.

For the three illustrations given in Fig. 172, the power delivered under each condition can be summarized as follows:

Condition	θ, deg	$\cos \theta$	$EI \cos \theta$
1	0	1.000	EI
2	36.9	0.800	0.8 EI
3	90	0.000	0

In practice the power factor, or $\cos \theta$, is determined from readings of wattmeter, voltmeter, and ammeter. Since

$$P = EI \cos \theta$$

then

$$\cos \quad P \tag{42}$$

For example, if a wattmeter indicates that a total power of 800 watts is passing through a circuit with a voltage of 100 and 10 amp flowing, the power factor is

$$\cos \theta = \frac{800}{100 \times 10} \quad 0.8$$

In practical calculations for power-factor improvements, it is necessary to determine the condenser capacity that must be added to the system in order to decrease the angularity between the voltage and current and increase the power factor of the load. For example, the load on a plant is 500 kw with a power factor of 0.71. It is desired to add sufficient condenser capacity to raise the power factor to 0.85. A graphical solution is shown in Fig. 173. Along the horizontal axis the load of 500 kw is drawn

to scale (AB). At the right end of the line AB a perpendicular is erected. By means of a protractor the angle of 44.8 deg cor-

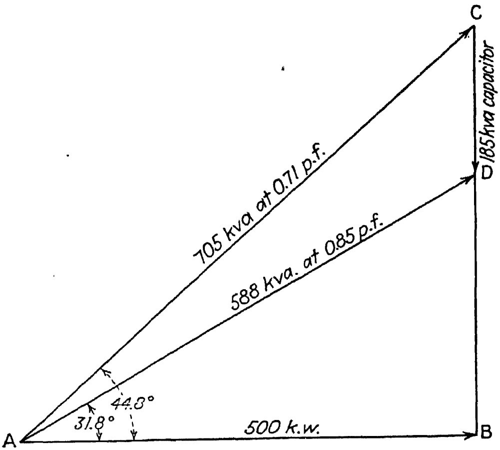

FIG. 173.—Graphical determination of size of capacitor required for power-factor correction.

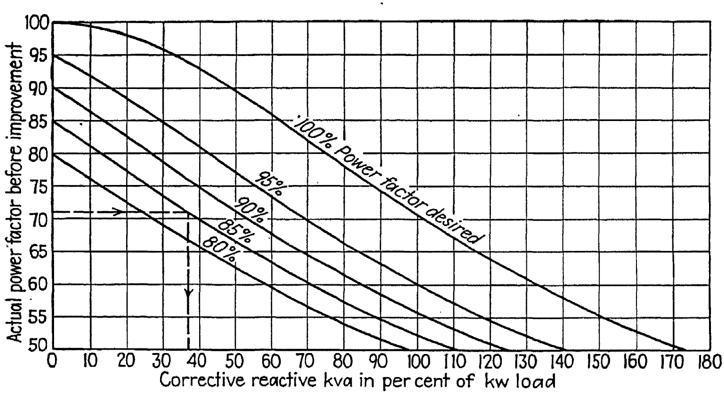

FIG. 174.—Chart for determining power-factor correction.

responding to $\cos \theta = 0.71$ is laid off with A as the center and AB as one side. The other side of the angle is projected until

it intersects the perpendicular at C. Likewise an angle of 31.8 deg corresponding to $\cos \theta = 0.85$ is plotted with A as a center and the line AD drawn. The distance DC, which is to the same scale as AB, represents the size of the capacitor needed, or 185 kva.

A shorter method for determining the size of a capacitor for any condition and for any desired power-factor improvement involves the use of Fig. 174 and a simple calculation. As in the example just given, for a power factor of 0.71 which it is desired to improve to 0.85, begin with a value on the left-hand vertical scale equal to the existing power factor and proceed horizontally to the right until the scale for the desired power factor is reached. From this point, proceed downward to the horizontal axis where a value of 37 per cent is read. This is the condenser capacity in kilovolt-amperes expressed as a percentage of the load in kilowatts. Thus, for a 500-kw load at 0.71 power factor the capacitor size necessary to improve the power factor to 0.85 would be $500 \times 0.37 = 185$ kva which is the same value as was determined graphically.

This series of curves can also be used to determine the power-factor correction that will be effected by a change in load and power factor on the system. For example, with a capacitor of 185 kva installed, consider that the load increases to 1,000 kw and that the power factor without the capacitor is 0.90. It is desired to find what improvement 185 kva of capacitor will make under these conditions. Since 185 is 18.5 per cent of 1,000, start on the horizontal axis at 18.5 per cent and proceed upward until the horizontal line for 0.90 power factor is reached. This indicates that the power factor will be raised to slightly over 0.95.

It is usual to locate condensers on the primary distribution feeders in order to improve the power factor of the alternating current flowing through the circuit. This has further beneficial effects insofar as the feeder circuit is concerned over and above the improvement of generator performance in the power plant. Where the generating plant is supplying an industrial establishment, the ideal arrangement is to provide capacitors at the power terminals of each motor. This cannot always be done economically, and it then becomes necessary to install the capacitors in locations on plant feeder circuits where the greatest benefit from the power-factor correction can be secured.

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CHAPTER XIX

ELECTRICAL SYSTEM LAYOUT

This chapter deals with electrical control and wiring essential in any electric generation station including layout of the station power bus, motors for driving plant accessories, auxiliary power supply, and wire and wiring accessories. Each item is considered from the standpoint of both suitability and reliability. Capacity limitations of the various items are discussed in detail where necessary to convey a thorough understanding of the workings of any particular item of equipment.

234. Station Power.—The electrical supply for operating cooling-water pumps, lubricating- and fuel-oil pumps, crane, centrifuges, electric heaters, and other motor-operated devices is usually 240 volts, three phase. With motors of sizes above 25 hp, it is usual to employ 480 volts, although under some conditions it may be economical to operate motors at 2,400 volts.

Station power is supplied through transformers fed from the main station power bus. It is advisable to install two station power banks, each being supplied by means of an independent circuit from the main power source. In such an arrangement, shown in Fig. 175, the normal power supply is provided by a transformer bank fed from a direct circuit on the 2,400-volt main power bus. The second, or emergency transformer bank, is fed through an auxiliary connection to one of the feeder circuits leaving the station. This arrangement is the least expensive that can be utilized since it eliminates the cost of a second 2,400-volt breaker and its necessary switchgear section for the emergency bank, while still providing a reasonably dependable auxiliary power supply in the event of failure of the main power bank or its supply.

On the low-voltage side of the two power banks, wiring connections are arranged in a manner to permit either or both to supply the station power system. In this instance the auxiliaries are fed from two panel boards and so divided between them that a

failure on any part of the 240-volt supply would affect only one-half of the plant auxiliaries. With this arrangement, power for essential circulating-water and lubricating-oil pumps is always available.

In laying out the station power supply it is first necessary to make up a list of all motor and other electrical loads which are to be supplied. This list is then divided into those items which

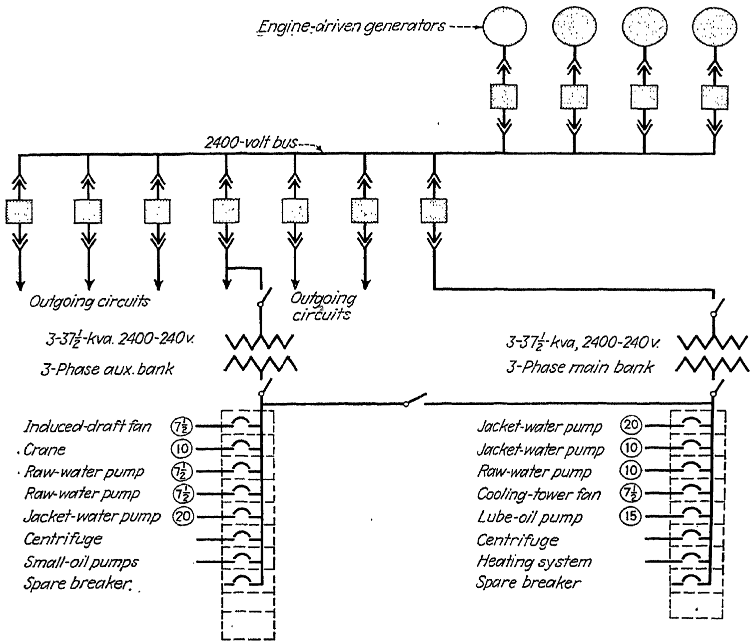
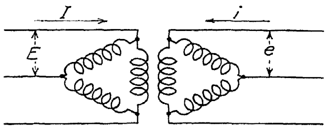
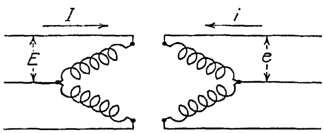
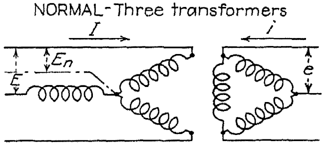
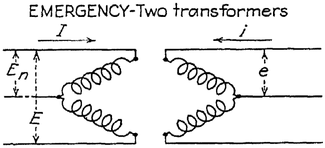


FIG. 175.—Station auxiliary power supply including main and emergency power banks. (*Courtesy of Power.*)

must operate at all times, such as circulating-water and lubricating-oil pump motors, and those items of equipment which operate intermittently or which can be out of commission for some time without affecting the operation of the plant. The items in continuous operating or preferred operating service should then be divided between the two panel boards in such a manner that the failure of one panel board or its power-supply

circuit will not interrupt plant service. Those items of equipment which operate intermittently can be divided between the two panel boards in any manner to balance the connected load between the two boards.

	Capacity of bank	High voltage side	Low voltage side
Δ-Δ CONNECTED BANKS NORMAL-Three transformers 	3 × K	$I = \frac{1732 \times K}{e}$	$i = \frac{1732 \times K}{e}$
EMERGENCY-Two transformers 	1.732 × K	$I = \frac{1000 \times K}{E}$	$i = \frac{1000 \times K}{e}$
Y-Δ CONNECTED BANKS NORMAL-Three transformers 	3 × K	$I = \frac{1732 \times K}{E}$ $I = \frac{1000 \times K}{E_n}$	$i = \frac{1732 \times K}{e}$
EMERGENCY-Two transformers 	1.732 × K	$I = \frac{1000 \times K}{E}$ $I = \frac{578 \times K}{E_n}$	$i = \frac{1000 \times K}{e}$

K=Capacity of one transformer in kva
Losses within the transformer have been neglected

FIG. 176.—Transformer connections and calculations.

235. Transformers.—Transformers supplying station power are generally single-phase units, since the failure of the windings in any one transformer will not necessarily put the entire bank out of commission. If three-phase transformers are used, the failure

of any part of the transformer windings would put either the main or emergency supply out of service until another transformer could be secured. If both the main and emergency power supply banks utilize single-phase transformers of the same capacity, one spare transformer will suffice as a replacement unit for either bank. Furthermore, by the use of single-phase units, it is possible to remove one of the three from service and still provide approximately 58 per cent of the capacity of the bank with the two remaining transformers. Wiring connections for station power banks served from 2,400- and 4,160-volt circuits supplying three-phase power under both normal and emergency conditions are shown in Fig. 176.

The size of the transformer bank for main and emergency supply must be determined from a study of the auxiliaries installed in the station, the capacity of motors and other equipment normally operating, and the probable maximum horsepower in motors and other load in kilowatts which must be served simultaneously. Any study made must assume certain operating conditions. In view of this fact, it is always advisable to be liberal in selecting the size of transformers for station power purposes. Even though transformers can be overloaded for relatively short periods of time, it is not advisable to rely on this overload capacity in the initial plant layout.

In determining the size of a station power bank, it is usual to figure 1 kva of transformer capacity will be required for each motor horsepower in operation. As an example of the method for calculating the capacity of a station power bank, the motor sizes and other loads shown in Fig. 175 will be used. A total of 62.5 hp is supplied from the main 240-volt panel while 52.5 hp is connected to the auxiliary panel, exclusive of centrifuge strip heaters and other small auxiliaries. If all motors on the main panel, together with a 5-kw heating element for lubricating-oil purifying, are being operated, the total load would approximate $62.5 + 5$, or 67.5 kva. The capacity of the main station transformer bank is 3×37.5, or 112.5 kva, which is considerably more than is required to supply the load. Using the next smaller standard size transformer shown in Table 63, or 25 kva per single-phase unit, would provide a capacity of only 75 kva per bank. This does not provide sufficient spare capacity above that necessary for present maximum requirements to allow

much addition in station power load. Had 75-kva banks been installed, the addition of a single 10-hp motor would have absorbed all of the spare transformer capacity. In reality, the addition of another generating unit will require a 7.5-hp motor for an induced-draft fan on the cooling tower, a 20-hp

FIG. 177.—Station power and lighting transformers installed in a separate vault.

jacket-water-pump motor, and a 10-hp raw-water-pump motor, or an additional load of 37.5 hp. The selection of 37.5-kva transformer units in this particular example was justified, therefore, when consideration is given to probable future additions.

The performance of standard transformers of the sizes generally used for station lighting and power service is given in Table 63 for 2,400-volt units.

TABLE 63.—PERFORMANCE OF 60-CYCLE SINGLE-PHASE DISTRIBUTION TRANSFORMERS; STANDARD VOLTAGES 2,400 TO 120/240 AND 240/480 VOLTS

Kva	Full load			
	Iron loss, watts	Copper loss, watts	Total loss, watts	Full load efficiency, per cent
1½	22	37	59	96.2
3	32	58	90	97.1
5	39	96	135	97.4
7½	52	136	188	97.5
10	57	188	245	97.6
15	77	261	338	97.8
25	115	388	503	98.0
37½	148	512	660	98.2
50	186	617	803	98.4
75	280	930	1,210	98.4
100	370	1,200	1,570	98.5

NOTE.—Reactance of transformers varies from 3 to 6 per cent depending upon design.

236. Overvoltage Taps.—Where an electric generating station is supplying a distribution system in a community, it is usually found advisable to maintain the voltage on the station primary bus above normal in order to compensate partially for voltage drop on primary circuits. If it is desired to obtain 2,300 volts at the outer end of radial feeders, it may be necessary to carry the station bus voltage as high as 2,500 volts during heavy load periods.

Transformers for power service have a fixed voltage-transformation ratio, and one designed to deliver 240 volts from a 2,400-volt supply would have a 10:1 ratio. If the primary voltage is raised to 2,500 volts, the voltage on the secondary side is also raised, in this instance to 250 volts. This secondary voltage may be excessive and cause trouble with magnetic contactors and motors designed for operation at 220 volts under many conditions. In order to reduce this voltage below 250 volts when the primary bus is maintained at 2,500 volts, it is necessary to provide some means for changing the transformer

ratio of the station power bank. This is done by incorporating overvoltage taps in the transformers. Such taps must be specified when purchasing the transformer. It is the usual practice to provide two 2.5 per cent taps or one 5 per cent overvoltage tap, although other taps may be obtained.

237. Protection of Low-voltage Circuits.—The protection of low-voltage circuits against short circuits is required by the National Electric Code. This protection can be accomplished either by the installation of suitable fuses in each circuit, or by the use of low-voltage air breakers which, incidentally, are rapidly supplanting fuses in many installations. Fuses are not satisfactory on three-phase circuits, since the failure of one fuse will cause motors to operate single-phase, thereby overheating and probably destroying the motor windings. Breakers, on the other hand, open all three phases and eliminate this difficulty.

While the protection required for single-phase lighting circuits is similar to that necessary for the protection of station power circuits, the transformer capacities involved do not usually require as great interrupting capacity as is necessary for power circuits.

The standard sizes of low-voltage breakers available for mounting in panel assemblies similar to that shown in Fig. 178 are given in Table 64. If required, larger air breakers can be secured with current-interrupting ratings of 20,000 to 80,000 amp. It is seldom that breakers of these larger interrupting capacities are necessary in internal-combustion-engine power plants.

There are two types of low-voltage breakers available for installation in panel boards: the multi-breaker type which is built in only the 50-amp. frame size, and the regular air breaker, similar to the deion, type AB, as manufactured by the Westinghouse Electric and Manufacturing Company or the air breaker, type AF, as manufactured by the General Electric Company. Cabinets for housing a group of breakers vary in size, depending upon the number, type, and voltage of the breakers used.

The cheapest panel-mounted board consists of a group of multi-breakers installed in a cabinet. In this type of arrangement the largest breaker that can be used is a 50-amp unit and all breakers used with this type of assembly employ the same frame size for all current capacities listed under the 50-amp

frame size in Table 64. Boards of this type are satisfactory for 115/230 volt single-phase service but they are not suitable for three-phase station power service. Their field is essentially

TABLE 64.—LOW-VOLTAGE BREAKERS FOR PANEL MOUNTING

Ampere frame size	Continuous ampere capacity	Interrupting capacity, amp	
		250 volt a-c or 125/250 volt d-c	600 volt a-c or 250 volt d-c
50	15	5,000	10,000
	20	5,000	10,000
	25	5,000	10,000
	35	5,000	10,000
	50	5,000	10,000
100	50	15,000	15,000
	70	15,000	15,000
	90	15,000	15,000
	100	15,000	15,000
225	70	15 000	15,000
	90	15,000	15,000
	100	15,000	15,000
	125	15,000	15,000
	150	15,000	15,000
	175	15,000	15,000
	200	15,000	15,000
	225	15,000	15,000
600	225	15,000	15,000
	250	15,000	15,000
	275	15,000	15,000
	300	15,000	15,000
	325	15,000	15,000
	350	15,000	15,000
	400	15,000	15.000
	450	15,000	15,000
	500	15,000	15,000
	550	15,000	15,000
	600	15,000	15,000

that of a distribution center for lighting. The mains provided in this type cabinet are available in 50,100, and 200 amp capacity only.

For three-phase power supply at 240 volts where moderate capacities are involved, a panel arrangement is available using 250-volt a-c breakers, either AB or AF types.

In order to provide a panel-board assembly that can be changed to accommodate increased circuit capacity, the convertible board has been developed. It will accommodate either

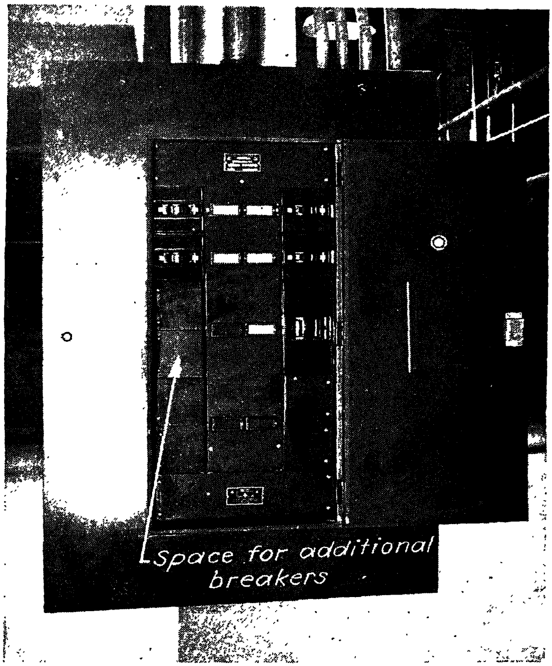


FIG. 178A.—Six hundred-volt convertible panel board serving plant auxiliaries.

250- or 600-volt a-c breakers. The mains provided in a panel board of this type can be 225, 400, 600, 800, or 1,200 amp capacity depending upon the power requirements that must be served. A convertible board can be installed to operate at 240 volts, and if necessary at a future date, it can be changed to serve as a 440-volt distribution center.

238. Selection of Low-voltage Breakers.—The size of an air breaker is determined by the normal current it must carry as well

as the short-circuit current that it may be required to interrupt. While the problem is somewhat comparable to that of selecting breakers for the control of circuits from the main power bus as given in Art. 229, Chap. XVIII, the calculations involved are much simpler.

Consider the station power supply shown in Fig. 175 having two banks, each consisting of three 37.5-kva single-phase trans-

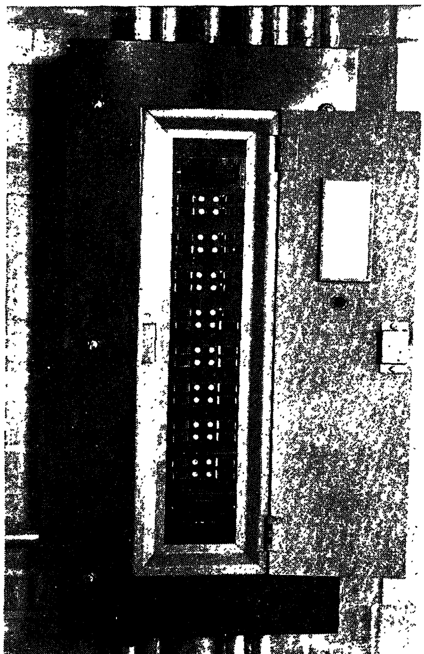


FIG. 178*B*.—Multi-breaker panel for controlling lighting circuits.

formers, or 112.5 kva per bank. In calculating the magnitude of short circuits on the low-voltage side of such a bank, it is advisable to assume that the high-voltage power supply is of infinite capacity and that regardless of the load imposed upon the low-voltage side of the transformer there will be sufficient capacity on the high-voltage side to supply it. Assuming a reactance of transformer and wiring of 5 per cent, the maximum

kilovolt-amperes that the transformer bank can supply is twenty times its full-load rating. From equations for a delta system, Fig. 176, the amperes per phase on the low-voltage side of the bank are $(1{,}732 \times 37.5)/240$, or 271 amp. When a short circuit occurs on the secondary side of a transformer fed from an infinite bus, the secondary voltage remains constant while the current increases in direct proportion to the short-circuit load. In other words, while the normal full-load current on the

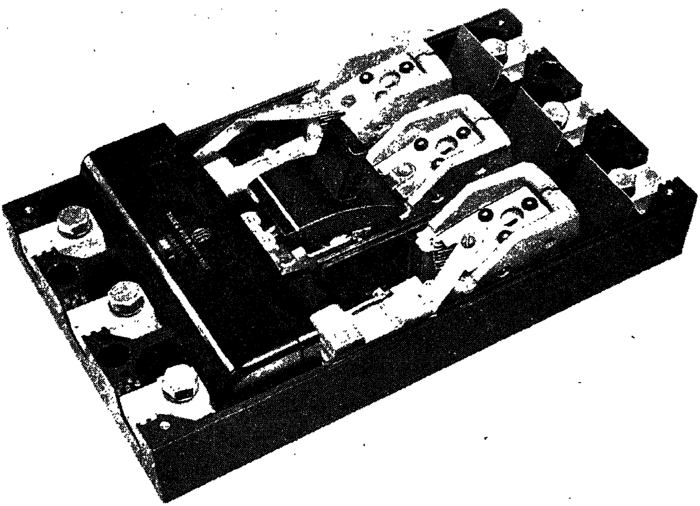


FIG. 178C.—Three-pole air circuit breaker for installation in convertible panel board. (*Courtesy of I-T-E.*)

secondary side of the transformer bank is 271 amp per phase, the short-circuit current is 20×271, or 5,420 amp per phase.

This requires the use of breakers having an interrupting capacity of 10,000 amp.

After the interrupting capacity has been determined, the next step in sizing the breaker is to select the proper current-carrying capacity. This requires that the full-load current of the motor be determined as well as the breaker size for carrying this current while allowing for motor starting. The full-load current values for three-phase a-c motors are set forth in Table 65.

TABLE 65.—FULL-LOAD CURRENT FOR INDUCTION-TYPE SQUIRREL-CAGE AND WOUND ROTOR MOTORS, AMPERES

Hp	Voltage of motor		
	110	220	440
½	5	2.5	1.3
¾	5.4	2.8	1.4
1	6.6	3.3	1.7
1½	9.4	4.7	2.4
2	12	6	3
3	....	9	4.5
5	....	15	7.5
7½	....	22	11
10	....	27	14
15	....	38	19
20	....	52	26
25	....	64	32
30	....	77	39
40	....	101	51
50	....	125	63

NOTE.—For additional values consult Table 24, p. 323, 1940 National Electric Code.

TABLE 66.—AIR CIRCUIT BREAKER SIZES FOR HIGH-REACTANCE AND AUTOTRANSFORMER STARTING THREE-PHASE SQUIRREL-CAGE MOTORS

Motor Full-load Current, Amperes	Breaker Size, Amperes
1– 6	15
7– 9	20
10– 12	25
13– 18	35
19– 30	50
31– 46	70
47– 60	90
61– 66	100
67– 90	125
91–110	150
111–129	175
130–150	200

NOTE.—For grouping small motors under the protection of a single circuit breaker, see Sec. 4343, 1940 National Electric Code.

Data contained in Table 66, based upon the 1940 National Electric Code, should be used in selecting the proper ampere rating of an air breaker for the protection of an electric motor circuit.

It is now possible to select the breakers necessary for the motor horsepower loads shown in Fig. 175. The top unit on the main panel is a 20-hp jacket-water-pump motor. Full-load current for this motor is 52 amp, Table 65, and according to Table 66, the breaker capacity should be 90 amp. Either a 250- or a 600-volt breaker in a 100-amp frame size has sufficient interrupting capacity. The 600-volt breaker was selected because it would permit conversion to 440-volt operation at a later date. The breaker sizes for all five motors shown on the main panel are summarized in Table 67.

TABLE 67.—BREAKERS REQUIRED FOR MOTORS ON MAIN PANEL SHOWN IN FIG. 175

Item	Motor hp	Breaker data		
		Ampere rating	Voltage	Frame size, amp
Jacket-water pump	20	90	600	100
Jacket-water pump	10	50	600	50
Raw-water pump	10	50	600	50
Induced-draft tower fan	7½	50	600	50
Lubricating-oil pump	15	70	600	100

239. Motors.—The horsepower capacity of motors is influenced by the permissible temperature rise in the motor windings. This in turn is dependent upon the type of insulating material used. Most motors are provided with class *A* insulation, discussed in Art. 217, Chap. XVIII, although class *B* insulating materials are employed in special cases.

Motors are generally grouped in two classes known as general-purpose and special-purpose motors. The special-purpose motors include splash- and drip-proof and totally enclosed units. All others are included in the general-purpose classification.

General-purpose motors are designed to operate at full load with a temperature rise of 40 C (104 F) in the motor windings, above a 40 C (104 F) ambient, and employ class *A* insulating

material. These motors are capable of handling 15 per cent overload at rated voltage and frequency with a greater temperature rise.

Splash-proof and drip-proof motors include all protected, semiprotected, drip-proof, splash-proof, and drip-proof-protected motors. Motors of this type, equipped with class *A* insulation, are designed to operate at full load with a temperature rise of 50 C (122 F) in the windings. In some special applications, they may be obtained with class *B* insulation, and the permissible temperature rise for such units is 70 C (158 F). Splash- and drip-proof motors carry no overload rating as do the general-purpose motors.

Totally enclosed motors include those classified as totally enclosed, totally enclosed fan-cooled, explosion-proof, waterproof, dust-tight, and submersible. Motors of this type having

Table 68.—N.E.M.A. Standard Horsepower Ratings and Synchronous Speed Range for A-c Motors

Motor hp	60-cycle motors		25-cycle motors	
	Maximum	Minimum	Maximum	Minimum
½	.....	.....	750	750
¾	1,200	1,200	750	750
1	1,800	1,200	1,500	750
1½	3,600	1,200	1,500	750
2	3,600	900	1,500	750
3	3,600	900	1,500	750
5	3,600	900	1,500	750
7½	3,600	900	1,500	750
10	3,600	600	1,500	500
15	3,600	600	1,500	500
20	3,600	600	1,500	500
25	1,800	600	1,500	500
30	1,800	600	1,500	500
40	1,800	600	1,500	500
50	1,200	600	750	500
60	1,200	600	750	500
75	1,200	600	750	500

Note.—The 3,600- and 1,500-rpm ratings apply to squirrel-cage motors only. Lower speed motors are available but have not been standardized as to frame sizes by N.E.M.A.

class *A* insulating material are designed to operate at full load with a temperature rise of 55 C (131 F), while those having class *B* insulation have a permissible temperature rise of 75 C (167 F). Totally enclosed motors carry no overload rating.

Occasionally a designer or plant operator specifies a splash-proof motor or a totally enclosed motor to operate with a 40 C temperature rise. When this is done, it is necessary to provide an oversize unit in order to comply with the temperature-rise limitations imposed. Special care should be taken in setting

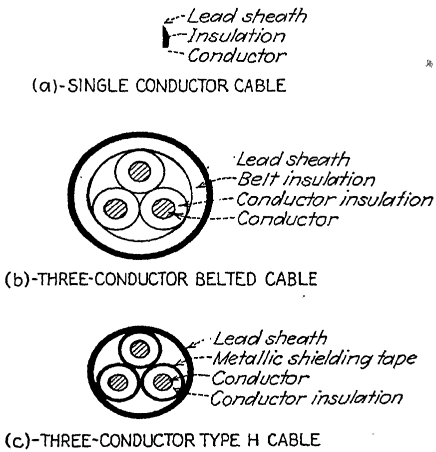


FIG. 179.—Types of insulated power cables.

temperature-rise limitations to see that standard machines are employed wherever possible.

240. Wire and Cable.—Numerous types of wire and cable are available for connecting electrical equipment with the power supply. Each type has inherent limitations from the standpoint of maximum permissible operating temperature, moisture conditions, operating voltage, cost, or ease of installation and splicing. It is necessary, therefore, in planning the plant wiring to have a knowledge of these factors in order that the proper type of wire be utilized for each particular service requirement.

Considerable development and improvement in the manufacture of paper, varnished cambric, asbestos, rubber, and synthetic insulated wire coupled with extensive research into

TABLE 69.—TYPES OF CONDUCTOR INSULATION FOR 600 VOLTS[1]

Trade name	Type name	Maximum operating temperature	Insulation	Thickness of insulation	Outer covering	Use
Code	R	50 C (122 F)	Code-grade rubber	14–10 . . . . . 3/64 8–2 . . . . . 4/64 1–4/0 . . . . . 5/64 250–500 . . . . . 6/64 501–1,000 . . . . . 7/64 Over 1,000 . . . . . 8/64	Moisture-resistant flame-retardant fibrous covering	General use
Moisture resistant	RW	50 C (122 F)	Moisture-resistant rubber	Same as type R	Moisture-resistant flame-retardant fibrous covering	General use or in wet locations
Performance	RP	60 C (140 F)	Performance-grade rubber	Same as type R	Moisture-resistant flame-retardant fibrous covering	General use
Heat resistant	RH	75 C (167 F)	Heat-resistant grade rubber	Same as type R	Moisture-resistant flame-retardant fibrous covering	General use
Small-diameter building wire (heat resistant)	RHT	75 C (167 F)	Heat-resistant grade rubber	14–10 . . . . . 2/64 8 . . . . . 3/64	Moisture-resistant flame-retardant fibrous covering	General use
Small-diameter building wire (performance)	RPT	60 C (140 F)	Performance-grade rubber	14–10 . . . . . 2/64	Moisture-resistant flame-retardant fibrous covering	Rewiring existing raceways
Type RU wire	RU	60 C (140 F)	90 per cent unmilled grainless rubber	14–10 . . . . . 18 mils	Moisture-resistant flame-retardant fibrous covering	Rewiring existing raceways
Solid synthetic	SN	60 C (140 F)	Solid flame-retardant moisture-resistant synthetic-compound	14–10 . . . . . 2/64 8 . . . . . 3/64 6–2 . . . . . 4/64 1–4/0 . . . . . 5/64	None	Rewiring existing raceways
Asbestos synthetic	SNA	90 C (194 F)	Synthetic and felted asbestos	14–8 . . . . . Synthetic . . . . . 20 mils Asbestos . . . . . 20 mils	Cotton braid thickness, 20 mils	Switchboard wiring

Varnished cambric	V	85 C (185 F)	Varnished cambric	Same as type R except 8 .... 3/64	Fibrous covering or lead sheath	Dry locations only unless lead sheathed. Smaller than No. 6 by special permission
Asbestos varnished cambric	AVA	110 C (230 F)	Impregnated asbestos and varnished cambric	14–8 sol. .... 45 mils 6–2 sol. .... 65 mils 1–4/0 sol. .... 75 mils 14–2 str. .... 60 mils 1–4/0 str. .... 80 mils 250–1,000 str. .... 110 mils	Asbestos braid	General use dry locations
Asbestos varnished cambric	AVB	90 C (194 F)	Same as type AVA	14–8 sol. .... 50 mils 6–4/0 sol. .... 80 mils 14–2 str. .... 60 mils 1–4/0 str. .... 80 mils 250–1,000 str. .... 110 mils	Flame-retardant cotton braid	General use dry locations
Asbestos varnished cambric	AVL	110 C (230 F)	Same as type AVA	14–2 .... 75 mils 1–4/0 .... 80 mils 250–1,000 .... 110 mils	Lead sheath	General use wet locations
Asbestos	A	200 C (392 F)	Felted asbestos	14–8 .... 40 mils 6–2 .... 60 mils 1–4/0 .... 90 mils 250–1,000 .... 120 mils	With or without asbestos braid	Dry locations only. Not for general conduit installation. In raceways, only as leads to or within apparatus. If without braid or moisture resistant treatment. Limited to 300 volts
Impregnated asbestos	AI	125 C (257 F)	Impregnated felted asbestos	Same as type A	With or without impregnated asbestos braid	
Paper		85 C (185 F)	Paper	....	Lead sheath	See National Electrical Code
Slow burning	SB	90 C (194 F)	3 braids, impregnated, fire-retardant thread	Same as type V	Outer cover finished smooth and hard	For use only in dry locations where the room temperature exceeds 85 C (185 F)
Slow burning, weatherproof	SBW	90 C (194 F)	2 layers impregnated cotton thread	Same as type V	Outer fire-retardant coating	For use only in dry locations where the room temperature exceeds 85 C (185 F)
Weatherproof	WP	80 C (176 F)	At least three cotton braids impregnated	14–12 .... 3/64 10–2 .... 4/64 1–4/0 .... 5/64 225–500 .... 6/64 525–950 .... 7/64 1,000 and over 8/64	....	May be used for interior wiring only by special permission

Courtesy of *Power*.
[1] Recognized in 1940 edition of the National Electrical Code.

TABLE 70.—CURRENT-CARRYING CAPACITY OF CONDUCTORS IN AMPERES PRESCRIBED BY THE 1940 NATIONAL ELECTRICAL CODE

Size AWG or MCM	Single conductor in free air; room temperature 86 F									Not more than three conductors in raceway or cable; room temperature 86 F							
	Rubber type R	Rubber type RP	Rubber type RHT type RH	Synthetic type SNA asbestos var-cam type AVB var-cam type V	Asbestos var-cam type AVA	Impregnated asbestos type AI	Asbestos type A	Slow-burning type SB weather-proof type W type SBW	Rubber type RW type R	Synthetic type SN type RU rubber type RPT type RP	Rubber type RHT type RH	Paper synthetic type SNA asbestos var-cam type AVB var-cam type V	Asbestos var-cam type AVA type AVL	Impregnated asbestos type AI	Asbestos type A
14	20	24	29	30	39	40	43	23	15	18	22	23	28	29	32
12	26	31	37	40	51	52	57	30	20	23	27	29	36	38	42
10	35	42	50	54	65	69	75	40	25	31	37	38	47	49	54
8	48	58	69	71	85	91	100	53	35	41	49	50	60	63	71
6	65	78	94	99	119	126	134	70	45	54	65	68	80	85	95
5	76	92	110	115	136	145	158	80	52	63	75	78	94	99	110
4	87	105	125	133	158	169	180	90	60	72	86	88	107	114	122
3	101	122	146	155	182	194	211	100	69	83	99	104	121	131	145
2	118	142	170	179	211	226	241	125	80	96	115	118	137	147	163
1	136	164	196	211	247	264	280	150	91	110	131	138	161	172	188
0	160	193	230	245	287	306	325	200	105	127	151	157	190	202	223
00	185	223	267	284	331	354	372	225	120	145	173	184	217	230	249
000	215	259	310	330	384	410	429	275	138	166	199	209	243	265	284
0000	248	298	358	383	446	476	510	325	160	193	230	237	275	308	340
250	280	338	403	427	495	528	562	350	177	213	255	272	315	334	372
300	310	373	446	480	555	592	632	400	198	238	285	299	347	380	415
350	350	421	504	529	612	653	698	450	216	260	311	325	392	419	462
400	380	457	547	575	665	710	755	500	233	281	336	361	418	450	488
500	430	517	620	660	765	814	870	600	265	319	382	404	468	498	554
600	480	577	691	738	857	912	970	680	293	353	422	453	525	543	612
700	525	632	756	813	942	1,003	1,065	760	320	385	461	488	562	598	668
750	545	655	785	846	981	1,044	1,118	800	330	398	475	502	582	621	690
800	565	680	815	879	1,020	1,085	1,150	840	340	410	490	514	600	641	720
900	605	728	872	941	.....	.....	.....	920	360	434	519	556			
1,000	650	782	936	1,001	1,163	1,238	1,332	1,000	377	455	543	583	681	730	811

TABLE 70.—CURRENT-CARRYING CAPACITY OF CONDUCTORS IN AMPERES PRESCRIBED BY THE 1940 NATIONAL ELECTRICAL CODE.—(*Continued*)

		Single conductor in free air; room temperature 86 F								Not more than three conductors in raceway or cable; room temperature 86 F						
Size AWG or MCM		Rubber type R	Rubber type RP	Rubber type RHT type RH	Syn-thetic type SNA asbestos var-cam type AVB var-cam type V	As-bestos var-cam type AVA	Impreg-nated asbestos type AI	Asbestos type A	Slow-burning type SB weather-proof type W type SBW	Rubber type RW type R	Synthe-tic type SN type RU rubber type RPT type RP	Rubber type RHT type RH	Paper synthetic type SNA asbestos var-cam type AVB var-cam type V	Asbestos var-cam type AVA type AVL	Impreg-nated asbestos type AI	Asbestos type A
1,250		740	890	1,066	1,131	.....	.....	.....	.....	409	493	589	643			
1,500		815	980	1,174	1,261	1,452	.....	.....	1,360	434	522	625	698	784		
1,750		890	1,070	1,282	1,370	.....	.....	.....	.....	451	544	650	733			
2,000		960	1,155	1,383	1,472	1,713	.....	.....	1,670	463	558	666	774	839		
		Correction Factor for Room Temperatures over 86 F														
C	F															
40	104	0.71	0.82	0.88	0.90	0.94	0.95	.....	.....	0.71	0.82	0.88	0.90	0.94	0.95	
45	113	0.50	0.71	0.82	.....	.....	.....	.....	.....	0.50	0.71	0.82				
50	122	0.00	0.58	0.75	0.80	0.87	0.89	.....	.....	0.00	0.58	0.75	0.80	0.87	0.89	
55	131	....	0.41	0.67	.....	.....	.....	.....	.....	....	0.41	0.67				
60	140	....	0.00	0.58	0.67	0.79	0.83	0.97	.....	....	0.00	0.58	0.67	0.79	0.83	0.97
70	158	....	.....	0.35	0.52	0.71	0.76	0.93	.....	....	....	0.35	0.52	0.71	0.76	0.93
75	167	....	.....	0.00	.....	.....	.....	.....	.....	....	....	0.00				
80	176	....	.....	.....	0.30	0.61	0.69	0.89	.....	....	....	....	0.30	0.61	0.69	0.89
90	194	....	.....	.....	.....	0.50	0.61	0.86	.....	....	....	....	....	0.50	0.61	0.86
100	212	....	.....	.....	.....	.....	0.51	0.82	.....	..	....	....	....	....	0.51	0.82
120	248	....	.....	....	.....	.....	.....	0.72	.....	....	....	....	....	....	....	0.72
140	284	....	.....	.....	.....	.....	.....	0.63	.....	....	....	....	....	....	....	0.63

Courtesy of *Power*.

the current-carrying capacities of the various types has resulted in an entirely new method for determining the ampere-carrying capacity of the various conductors. The 1940 edition of the National Electric Code sets forth the types of wire available and limitations on current-carrying capacities. The types of conductor insulation are shown in Table 69.

The current-carrying capacities for various conductor sizes insulated with these several materials are set forth in Table 70. These capacities are based upon an ambient air temperature of 30 C (86 F). For ambient temperatures exceeding this value, the current-carrying capacity of any wire is decreased according to the values in the lower portion of Table 70.

For example, if a conduit contains three No. 6 conductors insulated with type R or code-grade rubber having a maximum operating temperature of 50 C (122 F) each wire will safely carry 45 amp at an ambient temperature of 30 C (86 F). If the ambient temperature increases to 40 C (104 F), the current-carrying capacity of the wire decreases to 0.71 $\times$ 45, or 32, amp. Should the insulating compound be of a heat-resisting grade, type RH, for a maximum operating temperature of 75 C (167 F), the current-carrying capacity of the three No. 6 wires in conduit would be 65 amp when the ambient air temperature is 30 C (86 F) and 0.88 $\times$ 65, or 57.2, amp when the ambient increases to 40 C (104 F).

In examining Table 70, it becomes readily apparent that the current-carrying capacity does not increase so rapidly as the cross-sectional area of the conductor. Thus a No. 6 wire with type R insulation in free air has a current-carrying capacity of 65 amp, while a No. 3 wire, having double the cross-sectional area of the No. 6, can handle only 101 amp. Thus two No. 6 wires in parallel would have a capacity of 130 amp as compared with 101 amp for a No. 3 conductor of equal cross-sectional area.

There are many conditions, particularly when the current :ceeds 200 amp per phase, where the use of two or more con- :tors in parallel is necessary to keep the amount of copper within reasonable limits.

n addition to providing proper conductor size to carry the uired current it is necessary to have sufficient insulation kness for the particular voltage at which the conductor must

operate. Insulation thickness for 600-volt insulated wire is given in Table 69. Conductors operating at less than 600 volts use this same insulation thickness. The requirements for insulation thickness when using rubber, varnished cambric, and paper, operating at 600 volts and higher, are set by the standards of the Insulated Power Cable Engineers Association.

CHAPTER XX

EQUIPMENT TESTING

Machinery is tested to ascertain its capacity, efficiency, and other characteristics, or to determine whether or not it conforms to the guarantees under which it was purchased. Determination of capacity, efficiency, and general characteristics is usually considered as laboratory or development testing and will be considered only incidentally in this chapter. Commercial testing employed to determine whether or not an item of equipment complies with its guarantees will be considered in some detail.

241. Test Codes.—Standard codes of test procedure for mechanical and electrical equipment have been developed by the A.S.M.E., the A.I.E.E., the S.A.E., and other technical organizations interested in the standardization of test procedures. These recognized standards should be used for testing mechanical or electrical equipment wherever possible.

Before attempting to use any test code, ascertain its scope and methods of procedure as well as its limitations. These may include such things as variations in test loads, methods for conducting particular tests, or limits of application of the code in question. Many codes require that certain test procedures be determined by agreement between the manufacturer and purchaser. Where such conditions arise, it is advisable to settle all points which might be subject to dispute prior to the conduct of the test.

242. Physical Limitations of Tests.—It is not always possible to conduct tests strictly in accordance with the code, although every effort should be made to have tests conform with code requirements, even though some special effort may be necessary to obtain correct conditions. When deviations are necessary, it is always advisable for the interested parties to agree upon a suitable testing procedure before tests are conducted. Furthermore, when it does become necessary to deviate from the proce-

dure prescribed in a test code, the person responsible for making the tests should know that the deviations are such that a true indication of equipment performance and efficiency can be determined within reasonable limits.

243. Basis of Commercial Testing.—The commercial testing of any piece of equipment deals primarily with two objects, capacity and efficiency. It is necessary to know that a unit of sufficient size has been furnished. It is also necessary to ascertain whether or not the efficiency of the machine comes within the requirements of the purchase guarantee. Tests of equipment may include determinations of many other conditions, but the capacity and efficiency of the equipment are usually the primary objects.

Most test codes recognize this fact and are designed with this in view. If it is desired that the tests of any piece of equipment include other determinations, they should be set forth clearly in the specifications for the purchase of the equipment.

244. Accuracy and Precision.—The terms *accuracy* and *precision* are used in connection with test determinations, and it is advisable to know to what each term refers. Accuracy, according to the dictionary, means freedom from mistake and is usually applied to the human element in reading instruments. Accurate readings of an instrument or group of instruments are the actual values read and no errors are contained in the data, either in the actual reading of the instruments or in the recording of those readings. Precision, on the other hand, means exact limitation and refers to the correctness of adjustment of the measuring instruments.

In thinking of the two terms, it is advisable to consider them in the light of the tolerance involved. Accuracy has no tolerance, while precision includes a tolerance.

In general usage these two terms are employed synonymously, but in this chapter they will be employed in the manner in which they are here defined.

245. Precision of Instruments.—Instruments used for test purposes are not correct throughout the range of the instrument scale, and sometimes they are not correct at any point on the scale. This does not imply that instruments cannot be relied upon for making test determinations. There are tolerances which are permitted in the registering of any meter or instrument,

and the magnitude of these tolerances should be known in order to appraise any test information correctly. As an example, a voltmeter used on a power-plant switchboard, having a scale length over 5 in., is guaranteed to have a tolerance of ± 1 per cent of full-scale reading. If the scale of the instrument is calibrated from 0 to 300 volts, the meter may be in error as much as 3 volts and still be within the limits of precision as set by the A.S.A. Thus in reading a value of 157 volts on the meter, the actual voltage can lie anywhere between 154 and 160 volts.

Since tolerances of instruments must be taken into consideration if accurate test results are to be obtained, it is advisable to know the usual tolerances found in commercial test instruments. This information is summarized in Table 71.

246. Engine Testing.—Standard methods of procedure for the testing of internal-combustion engines have been prepared by the A.S.M.E., the S.A.E., and the United States Army and Navy. The Army and Navy requirements deal exclusively with acceptance tests for aircraft engines, while the test code of the S.A.E. is generally considered as being applicable to automotive and aircraft engines. The internal-combustion-engine test code developed by the A.S.M.E. is applicable to all types of internal-combustion engines.

It is highly desirable in any test to duplicate actual working conditions insofar as possible. The load supplied by the engine should be maintained constant at each test point, although this highly desirable condition is not always attainable. When the load does fluctuate, it is necessary to take sufficient load readings to establish the average load throughout the test period.

It was pointed out in Art. 66 that air temperature, air intake pressure, and exhaust back pressure all have a marked effect upon the capacity and fuel consumption of an engine. Careful measurements of the air temperature, pressure at the engine intake air header, and the pressure in the exhaust header should be made during the test runs. Such data are necessary in order to interpret properly the performance of the engine. Failure to obtain these data may lead to serious arguments between the engine manufacturer and purchaser, particularly in cases where contractual guarantees of efficiency and capacity have been limited by atmospheric temperature and pressure.

TABLE 71.—INITIAL TOLERANCE OF COMMERCIAL METERS AND INSTRUMENTS

Instrument	Standard	Tolerance			Notes
Barometric pressure:					
Aneroid barometer	Mfg.	0.01–0.1 in.			
Mercury barometer	Mfg.	0.001–0.01 in.			Must be corrected to 32 F
Dynamometers:					
Effective length torque arm	A.S.M.E.	0.2			Per cent } Set by internal-combustion-engine test code
Torque-arm force	A.S.M.E.	0.2			Per cent } Set by internal-combustion-engine test code
		Grade A	Grade B		
Electrical measurements:[a]					
Portable d-c instruments horizontal	A.S.A.	0.35	0.50		Per cent full-scale reading
Portable d-c instruments vertical	A.S.A.	0.70	1.00		Per cent full-scale reading
Portable a-c voltmeters	A.S.A.	0.30	0.80		Per cent full-scale reading
Portable a-c ammeters and milliammeters	A.S.A.	0.50	0.80		Per cent full-scale reading
Portable a-c single-phase wattmeters	A.S.A.	0.30	0.80		Per cent full-scale reading
Portable a-c polyphase wattmeters	A.S.A.	0.50	....		Per cent full-scale reading
		Scale length			
		Over 5 in.	Under 4.5 in.		
Switchboard a-c and d-c ammeters, voltmeters, and wattmeters	N.E.M.A. } A.S.A. }	1.0	2.0		Per cent full-scale reading
Switchboard frequency indicators, 60 cycles per second	A.S.A.	0.5	2.0		Per cent full-scale reading
Switchboard frequency indicators, 25 cycles per second	A.S.A.	1.0	2.0		Per cent full-scale reading
Power-factor instruments in terms of power factor	A.S.A.	1.0	2.0		Per cent full-scale reading
		Class			
		¼	½	1	
Instrument current transformers[b]	N.E.M.A.	0.25	0.50	1.00	Per cent
Instrument potential transformers	N.E.M.A.		0.50	1.00	Per cent

TABLE 71.—INITIAL TOLERANCE OF COMMERCIAL METERS AND INSTRUMENTS.—(*Continued*)

Instrument	Standard	Tolerance	Notes
Pressure measurements:			
Standard gauge	Mfg.	±1.0	Per cent of full-scale reading above first 5 per cent
Precision gauge	Mfg.	±0.5	Per cent full-scale reading above first 5 per cent
Test gauge	Mfg.	±0.5	Per cent of full-scale reading
Mercury U tube	....	....	Must be corrected for temperature
Water U tube	....	....	Must be corrected for temperature
Speed measurements:			
Speed counters	....	±0.3–1.0 per cent	Influenced by possible slippage and make of instrument
Stroboscopic counters	....	....	Very slight error
Temperature measurements:			
Pyrometer			
Millivoltmeter type	Mfg.	±1.00	Per cent of full-scale reading
Potentiometer type	Mfg.	±0.05	Per cent of full-scale reading
Thermometers, glass and mercury	....	1 F	Temperature range 32 to 300 F for thermometer completely immersed. May be as much as 7 to 10 F depending upon immersion depth and contact
Weight measurements:			
Beam scales	A.S.M.E.	±0.2	Per cent. Set by internal-combustion-engine test code

[a] KNOWLTON, ARCHER E.: "Standard Handbook for Electrical Engineers," 7th ed., p. 101, McGraw-Hill Book Company, Inc., New York, 1941.

[b] When switchboard instruments are used with either current or potential transformers, an additional precision tolerance of 1 per cent shall be added to the limit of precision for the instrument proper. When switchboard instruments are used with both kinds of transformers, the additional precision tolerance shall be 1.5 per cent.

Variations in air pressure, air temperature, and exhaust pressure are of vital importance because they all influence the weight of air delivered to the power cylinders for combustion of the fuel. Insufficient air causes incomplete burning of the fuel with a resultant loss of power and efficiency. This effect becomes noticeable in a diesel engine at approximately three-fourths load, is more pronounced at full load, and produces a very marked effect upon the overload capacity of the unit since the amount of air taken into the cylinder is constant at all engine loads.

247. Factory Testing of Engines.—The manufacturer prefers that tests be conducted on an engine at the factory where testing facilities are available for securing a complete test record and where any necessary adjustments can be made promptly. Factory tests are satisfactory, provided that field-installation conditions are reproduced. This requires the use of accessories and piping that will produce the same intake air and exhaust pressures encountered in the final installation, circulating-water temperatures comparable to those which will be used in the field, comparable lubrication, and other comparable conditions which may affect the engine performance.

Unless such precautions are taken, the manufacturer may be penalizing himself in that the factory performance may be inferior to the actual performance in the field. Such conditions have been found in the factory testing of engines. On the other hand, factory conditions may be unduly favorable to the engine, and when installed, the engine may not perform so well as it did in the shop owing to restrictions in air and exhaust lines, cooling-water temperature conditions, and other factors.

248. Field Testing of Engines.—Many engineers prefer to test internal-combustion engines after they have been installed in the plant and all adjustments have been made for actual operating conditions. There is much to be said in favor of field testing even though difficulties are experienced occasionally in arranging a suitable field test setup. Field tests permit the operating staff to see just what the unit will be under various loads, thereby helping to familiarize the staff with the new equipment and its limitations.

Since the engine is equipped with the air intake and exhaust system as well as other auxiliaries with which it will operate, it is

possible with field testing to check the performance of the completed assembly.

Both factory and field testing are employed, but for power plants the most satisfactory testing can usually be done in the field after the engine and all its auxiliaries are in operating order.

249. Liquid-fuel Consumption.—Liquid-fuel consumption of an internal-combustion engine is determined by weighing. Two methods are used, and either will give satisfactory results when properly applied. These are known as the short-time and long-time weight determinations. The short-time weight determination consists of ascertaining the length of time required for the engine to consume a definite weight of fuel, while the long-time weight determination consists of ascertaining the total weight of fuel consumed over an extended period of ½ to 2 hr or even longer.

The short-time fuel-weight determination requires the use of a sensitive platform scale, an oil container, and a stop watch or other suitable timing device. At the start of the test with the engine in operation, the oil container is filled and the weight set on the scale beam slightly less than the weight of the tank and oil. This results in the scale beam staying in its top position. At the instant the beam drops, the stop watch is started. After the beam has dropped, the weight on the beam is set for a predetermined number of pounds less than the weight at which the beam dropped. As soon as this new weight is set on the scale, the beam again rises and stays up until the required weight of oil is consumed. When the beam drops the second time, the watch is stopped, thereby obtaining the time required for the consumption of a definite weight of fuel.

If, for example, it is found that an engine delivering 300 hp required 5 min to consume 10 lb of oil, the oil consumption per horsepower-hour would be

$$\text{Fuel consumption} = \frac{60 \times 10}{5 \times 300} = 0.4 \text{ lb per bhp-hr}$$

When tests of this kind are made, it is advisable to repeat the test several times and take the average of the series. A more accurate arrangement using a mercury switch on the scale beam

for starting and stopping the timer is required by the A.S.M.E. Test Code for Internal-combustion Engines, No. 17.

The long-time fuel-weight determination requires the use of an accurate platform scale, an oil container, and an accurate clock or watch. For this type of test, the oil container should be equipped with a quick-closing valve for shutting off the tank periodically during refilling operations and for rapid closing at the termination of the test period, Fig. 180. At the start of the test period, the gross weight of the tank and oil is taken, and oil is allowed to flow from the tank to the engine until the tank is practically empty. When this condition occurs, the valve is closed, and the weight of the tank and remaining oil is taken. The tank is refilled and weighed, after which oil

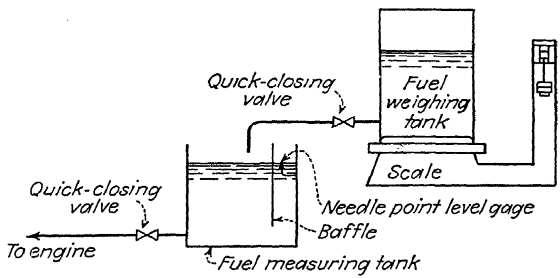


FIG. 180.—Arrangement for weighing fuel oil during engine test.

is again admitted to the engine. This process of weighing at the start and finish of each tank of oil consumed is continued throughout the test run. At the termination of the run, the oil flow is shut off by the quick-closing valve. The total weight of oil consumed is the sum of the differences in weight of each filled and empty tank used plus the difference between the filled and final weight of the last tank of oil used in the test.

When testing engines installed in the field, it is not always practical to set up an arrangement of this character. Equally good results can be obtained by the use of a platform scale on which is mounted an oil container from which fuel-oil suction is taken for the engine together with a second platform scale for weighing quantities of oil added to the container during the test run. At the start of the test, the weight of the tank and oil is taken. As oil is used from the tank on the first scale, weighed

quantities of oil are added. At the termination of the test the weight of the tank and oil contained therein is taken. The difference between the weight of the original tank of oil and the weight at the end of the test plus the weight of all oil added during the test period equals the total oil consumption.

250. Gas-fuel Consumption.—Measurements of gas consumption are best made by the use of a calibrated positive volumetric displacement meter or a drop holder. When the flow of gas is too great to permit the use of a displacement meter or drop holder, flow measurements employing a nozzle, orifice, or Venturi tube should be used. Whenever flow measurements are employed, extreme care must be exercised to ensure that correct

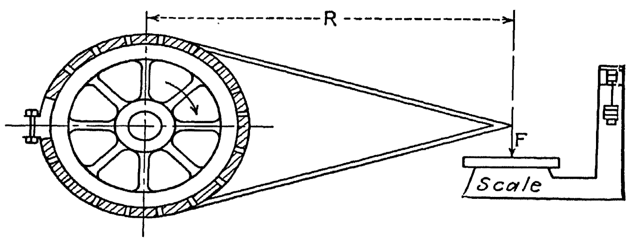


FIG. 181.—Prony brake for measuring engine horsepower output.

metering is obtained. The A.S.M.E. Code for Flow Measurements by Means of Standardized Nozzles and Orifice Plates and the A.G.A. Gas Measurement Committee Report No. 2 should be followed whenever flow measurements are used for determining the fuel consumption of a gas engine.

251. Determination of Engine Output.[1]—The shaft-horsepower output of an engine may be determined by the use of a suitable brake, dynamometer, or an electric generator with known efficiency characteristics. The simplest device for determining the horsepower output of an engine is the prony brake, Fig. 181, although this device is limited in the horsepower it can absorb. Hydraulic and electric dynamometers are used extensively for shop testing. If an engine drives an electric generator, the generator output, corrected for efficiency, can be used to determine the horsepower input. An electric generator is usually

[1] See A.S.M.E. Power Test Code No. 19, Part 7—Measurement of Power by Means of Dynamometers, 1941.

not satisfactory for loading an engine that is to operate over a wide speed range.

The prony brake is essentially a brake band to which is attached a lever arm. By clamping the brake band around the flywheel of the engine under test, a torque load can be produced at the end of the lever arm. This torque load is measured by a suitable platform scale. The brake band is tightened until an increase in load is shown by the scale, and this load is measured. The brake-horsepower output of the engine can then be computed by means of Eq. (43).

$$P_D = \frac{\qquad}{33,000} \tag{43}$$

where P_D = developed horsepower absorbed by the brake.
F = net force exerted by the free end of the brake arm on the weighing device, pounds.
R = distance from center line of rotating shaft to line of application of force at outer end of brake arm, feet.
n = revolutions per minute of engine flywheel.

It is usual to simplify this equation, since for a given brake the value $2\pi R/33,000$ is constant. The simplified equation is usually written in the form

$$P_D = \frac{Fn}{K} \tag{44}$$

where

$$K = \frac{33,000}{2\pi R}$$

and the other values are the same as those given for Eq. (43).

The hydraulic dynamometer works upon the same general principle as the prony brake, and Eqs. (43) and (44) are applicable. Instead of transmitting the torque to a platform scale, the hydraulic dynamometer employs a lever arm having a pan on which are placed known weights to counterbalance the dynamometer turning moment. Electric dynamometers in general operate in a manner similar to the hydraulic dynamometers with weights being added to the lever arm to counterbalance the engine torque.

252. Electrical Load Determination.—When the output of an engine is measured with an electric generator, special precautions

must be taken to ensure that the true efficiency of the generator for various loads is known and that the electrical instruments used for testing are accurate. *Furthermore, the person making the tests must be certain that he is reading the true power output of the generator.* Electrical circuits must be checked, and all corrections for meters, instrument transformers, and other possible errors determined and applied. True power can be obtained either by means of single-phase wattmeters, or by

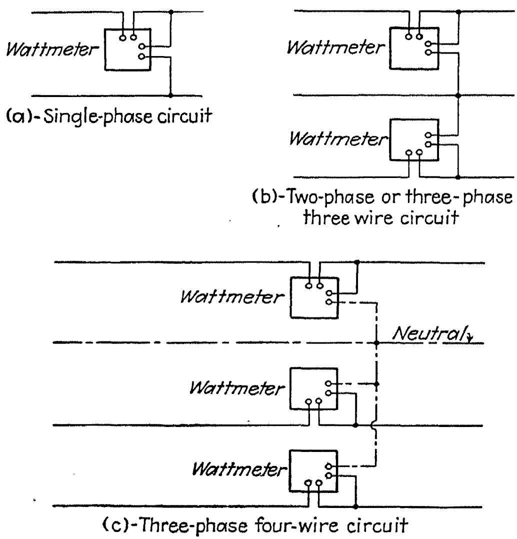


FIG. 182.—Wattmeter connections for measuring electrical-power output.

determining the speed of rotation of the polyphase watthour-meter disks.

Determination of the electrical load with single-phase wattmeters is a recognized standard method of electrical load determination. For a single-phase generator, only one single-phase wattmeter is required. For two-phase and three-phase delta-connected generators, as well as three-phase wye-connected generators where the neutral is not brought out, two single-phase wattmeters are required to measure the total power. Where three-phase, wye-connected generators have the neutral brought out, it is

necessary to use three single-phase wattmeters for measuring the true power output. Where two or three single-phase wattmeters are used, it is necessary to read all meters simultaneously and take the sum of all wattmeter readings as the total power delivered. Where current and potential transformers are used in conjunction with the wattmeters, it is necessary to determine the correct multipliers for these instrument transformers in order to determine the true power output. Wattmeter connections for measuring power in various circuits are shown in Fig. 182.

Determination of the electrical load with a polyphase watthour meter is readily accomplished by ascertaining the speed of rotation of the meter disks. Every polyphase watthour meter is provided with an indicating point or stripe on each rotating element for assisting in counting revolutions. With the aid of a stop watch, the time for 10 revolutions of the watthour-meter disks can be determined, and by substituting in Eq. (45) the load in kilowatts can be computed.

$$3.6K_h \tag{45}$$

where P_b = measured power output, kilowatts.
K_h = meter constant or watts per revolution of the meter disk.
t_w = time for one revolution of the meter disk, seconds.

Since polyphase watthour meters can be readily calibrated in the field with the aid of a rotating standard, and since the timing of several revolutions of the meter disk with a stop watch gives a very accurate value of the speed of rotation of the meter disk, this method for checking generator loads is preferred by many engineers.

253. Electrical Generator Performance.—Tests of electrical generators are usually made at the factory in conformity with the Standards of the A.I.E.E. It is somewhat difficult to run tests on an electric generator in the field. Factory tests include high potential tests of insulation, resistance of windings, heat runs to determine the temperature rise in the windings, and efficiency tests. As pointed out in Art. 216, the conventional A.I.E.E. efficiency of a generator does not include exciter and rheostatic

losses, and these must be included to determine the true over-all efficiency of the generating unit.

Whenever an engine-generator unit is being tested in the field, it is customary to make spot checks of the winding and bearing temperatures of the generator to supplement the factory tests on the machine. This ensures that no damage has been done to bearings or windings following the shop tests.

254. Cooling-tower Tests.—The cooling tower is usually tested at the same time the engine-generator units are being field tested. In order to check the performance of the cooling tower, it is necessary to know the wet- and dry-bulb temperatures as well as the quantity of water being passed over the tower. A sling psychrometer is usually employed to determine the wet-bulb temperature.

255. Cooling-system Tests.—It is advisable to make a thorough inspection of the entire cooling-water piping system in the plant to ensure that all parts of it are functioning correctly before any plant testing is done. This should include checking the tightness of the system by means of a hydrostatic pressure test, checking the location and operation of all valves, performance of cooling-water pumps, and the cleanliness of all lines.

The capacity of a pump can be approximated by taking the pressure drop across the pump, and from the pump test curve for this pressure value, read the pumping rate. The pump capacity can also be determined by installing an orifice plate in the pump discharge line, and from readings of pressure differential across the orifice plate, the flow rate can be computed.

256. Fuel-oil System.—It is the usual practice to test the fuel-oil tanks, pumps, and piping system for oil tightness prior to filling them by the use of a suitable hydrostatic test. After the water pressure has been applied to the system, all water must be drained out before oil is placed in the storage tanks. After the oil has been put in the system, all joints should again be inspected for possible leaks which might have been overlooked during the hydrostatic test. The operation of all valves and pumps should be checked to ensure that they are functioning correctly.

257. Miscellaneous Tests.—Many miscellaneous items of equipment are used in any power plant, and these should be inspected thoroughly and operated sufficiently to ensure their

correct functioning after the plant is placed in operation. Electrical switchboard instruments and meters should be gone over thoroughly to ensure that all packing and blocking materials installed to protect the equipment during shipment are removed, and that the instruments are capable of operating before they are placed in service. Pressure gauges, thermometers, pyrometers, and all other instruments should likewise be checked to ensure that they are in operating condition. Such attention to detail, trivial as it might seem, will add much to the ease with which the new plant is put into operation.

Air compressors should be thoroughly checked before being started up. Particular attention should be given to the functioning of the safety valve on the compressor discharge line. Compressors have been damaged beyond repair through excessive air pressure resulting from the failure of a relief valve to open at the correct pressure. The automatic pressure-control switch for starting and stopping the motor-operated air compressor should also be inspected to make certain that it will stop as well as start up the compressor at the required pressure value.

APPENDIX

The following conversion tables for the basic English units of length, area, volume, weight, velocity, rate of flow, unit weight, pressure, energy, and power are reproduced by the courtesy of *Power*.

Any listed factor multiplied by a quantity expressed in the unit printed at the left on the same line will give the equivalent quantity in the unit heading the column.

Where L or R, with a number, is printed immediately below the factor, move the decimal point *left* or *right*, respectively, the indicated number of places.

All factors are carried to five significant figures (except where the factor is exact with a smaller number). One or two figures at the end may be dropped where slide-rule accuracy is sufficient.

WEIGHT

	Grains	Ounces	Pounds	Tons (short)	Tons (long)
Grains		0.22858 L 2	0.14286 L 3	0.71428 L 7	0.63775 L 7
Ounces	437.5		0.0625	0.3125 L 4	0.27902 L 4
Pounds	7,000	16		0.5 L 3	0.44643 L 3
Tons (short)	14,000 R 3	32,000	2,000		0.89286
Tons (long)	15,680 R 3	35,840	2,240	1.12	

VOLUME

	Cubic inches	Gallons	Cubic feet	Cubic yards	Acre feet
Cubic inches		0.43290 L 2	0.57870 L 3	0.21433 L 4	
Gallons	231		0.13368	0.49511 L 2	0.30689 L 5
Cubic feet	1,728	7.4805		0.37037 L 1	0.22957 L 4
Cubic yards	46,656	201.97	27		0.61983 L 3
Acre feet		32,585 R 1	43,560	1,613.3	

AREA

	Circular mils	Square inches	Square feet	Square yards	Acres	Square miles
Circular mils		0.78540 L 6				
Square inches	12,732 R 2		0.69444 L 2	0.77160 L 3	0.15942 L 6	0.24909 L 9
Square feet		144		0.11111	0.22957 L 4	0.35870 L 7
Square yards		1,296	9		0.20661 L 3	0.32283 L 6
Acres		62,726 R 2	43,560	4,840		0.15625 L 2
Square miles		40,145 R 5	27,878 R 3	30,976 R 2	640	

Velocity

	Inches per second	Inches per minute	Inches per hour	Feet per second	Feet per minute	Feet per hour	Miles per minute	Miles per hour
Inches per second.		60	3,600	0.83333 L 1		300	0.94697 L 3	0.56818 L 1
Inches per minute.	0.16667 L 1		60	0.13889 L 2	0.83333 L 1	5	0.15783 L 4	0.94697 L 3
Inches per hour.	0.27778 L 3	0.16667 L 1		0.23148 L 4	0 13889 L 2	0.83333 L 1		0.15783 L 4
Feet per second..	12	720	43,200		60	3,600	0.11364 L 1	0.68182
Feet per minute..	0.2	12	720	0.16667 L 1		60	0.18939 L 3	0.11364 L 1
Feet per hour.	0.33333 L 2	0.2	12	0.27778 L 3	0.16667 L 1		0.31566 L 5	0.18939 L 3
Miles per minute.	1,056	63,360		88	5,280	31,680 R 1		60
Miles per hour.	17.6	1,056	63,360	1.4667	88	5,280	0.16667 L 1	

Density

Tons are 2,000 lb; gallons are U.S. (231 cu in.)

	Ounces per cubic inch	Ounces per cubic foot	Pounds per cubic inch	Pounds per cubic foot	Pounds per cubic yard	Tons per cubic foot	Tons per cubic yard	Pounds per gallon
Ounces per cubic inch		1,728	0.06250	108	2,916	0.054	1.458	14.438
Ounces per cubic foot	0.57870 L 3		0.36169 L 4	0.0625	1.6875	0.3125 L 4	0.84375 L 3	0.83549 L 2
Pounds per cubic inch	16	27,648		1,728	46,656	0.864	23.328	231
Pounds per cubic foot	0.92592 L 2	16	0.57870 L 3		27	0.0005	0.0135	0.13368
Pounds per cubic yard	0.34293 L 3	0.59259	0.21433 L 4	0.37037 L 1		0.18519 L 4	0.0005	0.49512 L 2
Tons per cubic foot.	18.518	32,000	1.1574	2,000	54,000		27	267.36
Tons per cubic yard..	0.68585	1,185.2	0.42866 L 1	74.074	2,000	0.37037 L 1		9.9024
Pounds per gallon....	0.69264 L 1	119.69	0.43290 L 2	7.4805	201.97	0.37403 L 2	0.10099	

Pressure

	Inches water (60 F)	Feet water (60 F)	Inches Hg (62 F)	Feet Hg (62 F)	Ounces per square inch	Ounces per square foot	Pounds per square inch	Pounds per square foot
Inches water (60 F)		0.83333 L 1	0.73681 L 1	0.61401 L 2	0.57740	83.146	0.36088 L 1	5.1967
Feet water (60 F)	12		0.88417	0.73681 L 1	6.9288	997.75	0.43306	62.360
Inches mercury (62 F)	13.572	1.1310		0.83333 L 1	7.8370	1,128.5	0.48981	70.532
Feet mercury (62 F)	162.86	13.572	12		94.044	13,542	5.8777	846.38
Ounces per square inch	1.7319	0.14433	0.12760	0.10633 L 1		144	0.625 L 1	90
Ounces per square foot	0.12027 L 1	0.10023 L 2	0.88613 L 3	0.73844 L 4	0.69444 L 2		0.43403 L 3	0.625 L 1
Pounds per square inch	27.710	2.3090	2.0416	0.17013	16	2,304		144
Pounds per square foot	0.19243	0.16036 L 1	0.14178 L 1	0.11815 L 2	0.11111	16	0.69444 L 2	

WORK—ENERGY—HEAT
Bases: 1 kwhr = 3411.5 Btu, and 1 Btu = 778.57 ft-lb

	Btu	Inch-pounds	Foot-pounds	Watt-hours	Kilo-watt-hours	Kilo-watt-days	Kilo-watt-years	Horse-power-hours	Horse-power-days	Horse-power-years
Btu		9,342.7	778.57	0.29313	0.29313 L 3	0.12214 L 4	0.33461 L 7	0.39322 L 3	0.16384 L 4	0.44886 L 7
Inch-pounds	0.10703 L 3		0.83333 L 1	0.31374 L 4	0.31374 L 7	0.13073 L 8	0.35812 L 11	0.42088 L 7	0.17536 L 8	0.48043 L 11
Foot-pounds	0.12844 L 2	12		0.37649 L 3	0.37649 L 6	0.15687 L 7	0.42975 L 10	0.50505 L 6	0.21044 L 7	0.57651 L 10
Watthours	3.4115	31,873	2,656.1		0.001	0.41667 L 4	0.11415 L 6	0.13415 L 2	0.55896 L 4	0.15313 L 6
Kilowatt-hours	3,411.5	31,873 R 3	26,561 R 2	1,000		1.41667 L 1	0.11415 L 1	1.3415	0.55896 L 1	0.15313 L 3
Kilowatt-days	81,876	76,495 R 4	63,746 R 3	24,000	24		0.27397 L 2	32.196	1.3415	0.36753 L 2
Kilowatt-years	29,885 R 3	27,920 R 7	23,267 R 6	87,600 R 2	8,760	365		11,752	489.65	1.3415
Horsepower-hours	2,543.1	23,760 R 3	198 R 4	745.45	0.74545	0.31061 L 1	0.85093 L 4		0.41667 L 1	0.11415 L 3
Horsepower-days	61,034	57,024 R 4	47,520 R 3	17,891	0.17891	0.74545	0.20423 L 2	24		0.27397 L 2
Horsepower-years	22,278 R 3	20,814 R 7	17,345 R 6	65,301 R 2	6,530.1	272.09	0.74545	8,760	365	

Power

Bases: 1 kilowatt-hour = 3411.5 Btu and 1 Btu = 778.57 ft-lb

	Watts	Kilowatts	Horsepower	Foot-pounds per second	Foot-pounds per minute	Foot-pounds per hour	Btu per second	Btu per minute	Btu per hour	Btu per day	Btu per year
Watts		0.001	0.13415 R 2	73,781	44.268	2,656.1	0.94765 L 3	0.56859 L 1	3.4115	81.876	29,885
Kilowatts	1,000		1.3415	737.81	44,268	26,561 R 2	0.94765	56.859	3,411.5	81,876	29,885 R 3
Horsepower	745.45	0.74545		550	33,000	198 R 4	0.70642	42.384	2,543.1	61,034	22,278 R 3
Foot-pounds per second	1.3554	0.13554 L 2	0.18181 L 2		60	3,600	0.12844 L 2	0.77064 L 1	4.6238	110.97	40,505
Foot-pounds per minute	0.22589 L 1	0.22589 L 4	0.30302 L 4	0.16667 L 1		60	0.21407 L 4	0.12844 L 2	0.77064 L 1	1.8495	675.08
Foot-pounds per hour	0.37649 L 3	0.37649 L 6	0.50503 L 6	0.27778 L 3	0.16667 L 1		0.35681 L 6	0.21407 L 4	0.12844 L 2	0.30826 L 1	11.251
Btu per second	1,055.3	1.0553	1.4156	778.57	46,714	28,029 R 2		60	3,600	86,400	31,536 R 3
Btu per minute	17.588	0.17588 L 1	0.23593 L 1	12.976	778.57	46,714	0.16667 L 1		60	1,440	52,560 R 1
Btu per hour	0.29314	0.29313 L 3	0.39322 L 3	0.21627	12.976	778.57	0.27778 L 3	0.16667 L 1		24	8,760
Btu per day	0.12214 L 1	0.12214 L 4	0.16384 L 4	0.90112 L 2	0.54067	32.441	0.11574 L 4	0.69444 L 3	0.41667 L 1		365
Btu per year	0.33461 L 4	0.33461 L 7	0.44886 L 7	0.24688 L 4	0.14813 L 2	0.88873 L 1	0.31710 L 7	0.19026 L 5	0.11415 L 3	0.27397 L 2	

VOLUME RATES

Month used is exactly one-twelfth year

	Cubic feet per second	Cubic feet per minute	Cubic feet per hour	Cubic feet per day	Cubic feet per month	Cubic feet per year	Gallons per second	Gallons per minute	Gallons per hour	Gallons per day	Gallons per month	Gallons per year
Cubic feet per second		60	3,600	86,400	26,280 R 2	31,536 R 3	7.4805	448.83	26,930	64,632 R 1	19,659 R 3	23,591 R 4
Cubic foot per minute	0.16667 L 1		60	1,440	43,800	52,560 R 1	0.12468	7.4805	448.83	10,772	32,765 R 1	39,318 R 2
Cubic feet per hour	0.27778 L 3	0.16667 L 1		24	730	8,760	0.20779 L 2	0.12468	7.4805	179.53	5,460.8	65,529
Cubic feet per day	0.11574 L 4	0.69444 L 3	0.41667 L 1		30.417	365	0.86579 L 4	0.51948 L 2	0.31169	7.4805	227.53	2,730.4
Cubic feet per month	0.38052 L 6	0.22831 L 4	0.13699 L 2	0.32877 L 1		12	0.28465 L 5	0.17079 L 3	0.10248 L 1	0.24594	7.4805	89.766
Cubic feet per year	0.31710 L 7	0.19026 L 5	0.11415 L 3	0.27397 L 2	0.83333 L 1		0.23721 L 6	0.14232 L 4	0.85390 L 3	0.20494 L 1	0.62338	7.4805
Gallons per second	0.13368	8.0208	481.25	11,550	35,131 R 1	42,158 R 2		60	3,600	86,400	26,280 R 2	31,536 R 3
Gallons per minute	0.22280 L 2	0.13368	8.0208	192.50	5,855.2	70,262	0.16667 L 1		60	1.440	43,800	52,560 R 1
Gallons per hour	0.37133 L 4	0.22280 L 2	0.13368	3.2083	97.586	1,170.4	0.27778 L 3	0.16667 L 1		24	730	8,760
Gallons per day	0.15472 L 5	0.92833 L 4	0.55700 L 2	0.13368	4.0661	48.793	0.15574 L 4	0.69444 L 3	0.41667 L 1		30.417	365
Gallons per month	0.50868 L 7	0.30520 L 5	0.18313 L 3	0.43950 L 2	0.13368	1.6042	0.38052 L 6	0.22831 L 4	0.13699 L 2	0.32877 L 1		12
Gallons per year	0.42390 L 8	0.25434 L 6	0.15260 L 4	0.36624 L 3	0.11139 L 1	0.13368	0.31710 L 7	0.19026 L 5	0.11415 L 3	0.27397 L 2	0.83333 L 1	

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